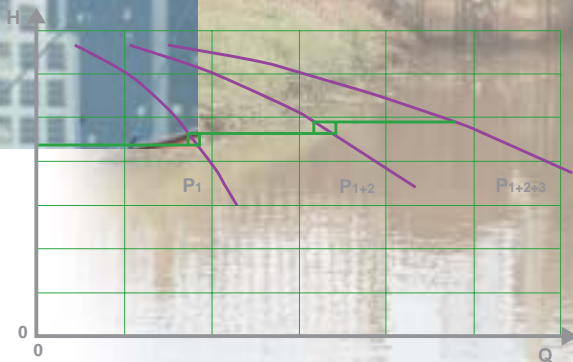


# Variable Speed Driven Pumps

## Best Practice Guide



AUTOMATION  
INSTRUMENTATION & CONTROL  
LABORATORY TECHNOLOGY



ACTIONenergy



# BPMA

*The Pump Industry Association*

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## **PREFACE AND ACKNOWLEDGEMENTS**

This guide is the result of co-operation between three different industries whose goal was to produce a document that would clearly define; in simple terms the information required when planning to use an electronic Variable Speed Driven Pumping System. The guide focuses mainly on applications within the Industrial Sector, however the principles used will be applicable to most pumping applications.

Members from the British Pump Manufacturers' Association (BPMA), Gambica's Variable Speed Drive group and experts from the Electric Motor industry assisted with this guide. The chairman of the working group is indebted to all the team members for their contributions.

In particular a special mention should be given to: -

**Geoff Brown – ABB Ltd., Drives & Motors**      **Darren Henighan – Brook Crompton Motors**

**Ian James – HMD Sealless Pumps**                      **John Bower – Flowserve**

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Sponsorship has been provided by Action Energy (formerly the Energy Efficiency Best Practice Programme). This continued support for energy saving equipment ensures that documents like this can be published.

To ensure the guide achieved its goal, copies were circulated to Technical Experts outside the working group who offered comment and direction. The entire working group are grateful for this extra guidance.

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**Steve Moore – Ineos Chlor**

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**Brian Nesbitt- Pump Consultant**

**Steve Schofield**  
**Director Technical Services BPMA**  
**(Chairman of the VSD working group)**

## **SECTION 1 INTRODUCTION**

Pump systems are often operated inefficiently. The reasons will vary from process to process and application to application, but the constant outcome is the cost to industry through wasted energy, which runs into millions of pounds per year, and the cost to the environment through the generation of this wasted energy.

It is estimated that in the United Kingdom, pumps use a total of 20TWh/annum, responsible for the emission of 2.7MtC/annum (2.7 million tons of carbon). Pumps therefore represent the largest single use of motive power in industry and commerce as shown in the breakdown of energy usage by motor driven equipment:

**Pumps -31%**

**Fans - 23%**

**Air Compressors - 8%**

**Other Compressors – 14%**

**Conveyors – 8%**

**Others 16%**

A pump installation is often sized to cope with a maximum predicted flow, which, may never happen. This principle of over sizing is frequently used in Industry, which subsequently leads to wasted energy and damage to parts of the pump installation.

Procurement costs of the pump equipment in general amount to less than 1% of the total investment of a plant, yet the operational quality of a pump may be the decisive factor in the overall functionality of the plant and its associated running costs.

Flow control by speed regulation of pumps, is one of today's best methods of varying the output on both Rotodynamic and Positive Displacement pumps and this guide describes its many advantages and potential system drawbacks.

The benefits covered include:

- **Energy cost savings**
- **Reliability improvements**
- **Simplified pipe systems (elimination of control valves & by-pass lines)**
- **Soft start & stop**
- **Reduced maintenance**

All amounting to lower life cycle costs.

Whilst other methods of control are available, this guide concentrates on Pulse Width Modulated Variable Speed Drive because it has the greatest benefits of control, energy efficiency, and ease of retrofitting.

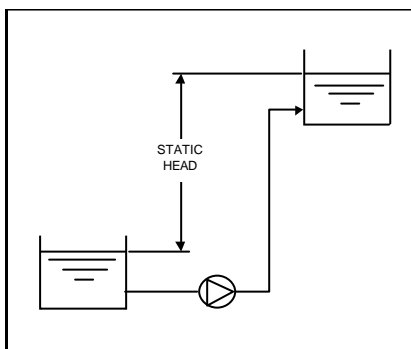
## **SECTION 2 PUMPING SYSTEM HYDRAULIC CHARACTERISTICS**

### **2.1 SYSTEM CHARACTERISTICS**

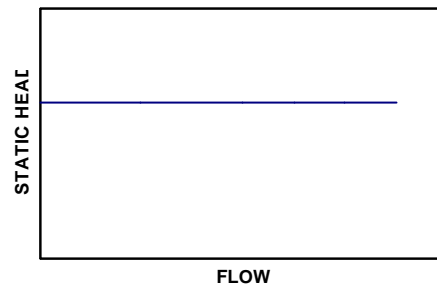
In a pumping system, the objective, in most cases, is either to transfer a liquid from a source to a required destination, e.g. filling a high level reservoir, or to circulate liquid around a system, e.g. as a means of heat transfer.

A pressure is needed to make the liquid flow at the required rate and this must overcome head 'losses' in the system. Losses are of two types: static and friction head.

Static head is simply the difference in height of the supply and destination reservoirs, as in Figure 2.1. In this illustration, flow velocity in the pipe is assumed to be very small. Another example of a system with only static head is pumping into a pressurised vessel with short pipe runs. Static head is independent of flow and graphically would be shown as in Figure 2.2.

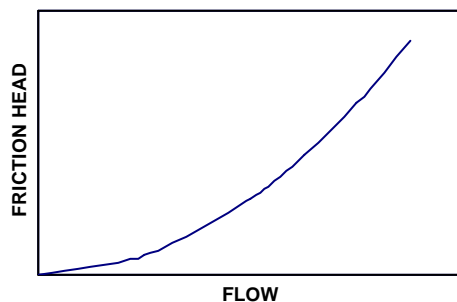


**Figure 2.1**  
**Static head**



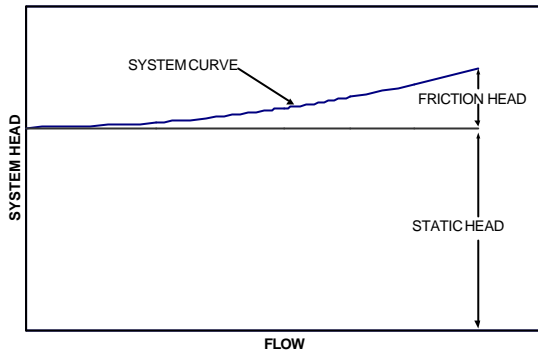
**Figure 2.2**  
**Static head vs. flow**

Friction head (sometimes called dynamic head loss) is the friction loss, on the liquid being moved, in pipes, valves and equipment in the system. The losses through these are proportional to the square of the flow rate. A closed loop circulating system without a surface open to atmospheric pressure, would exhibit only friction losses and would have a system head loss vs. flow characteristic curve as Figure 2.3

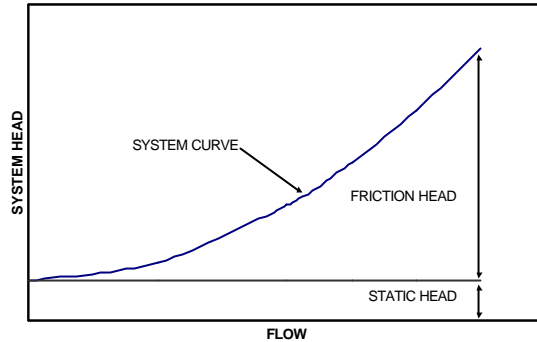


**Figure 2.3**  
**Friction head vs. flow**

Most systems have a combination of static and friction head and the system curves for two cases are shown in Figures 2.4 and 2.5. The ratio of static to friction head over the operating range influences the benefits achievable from variable speed drives (see section 3.2.2)



**Figure 2.4**  
System with high static head

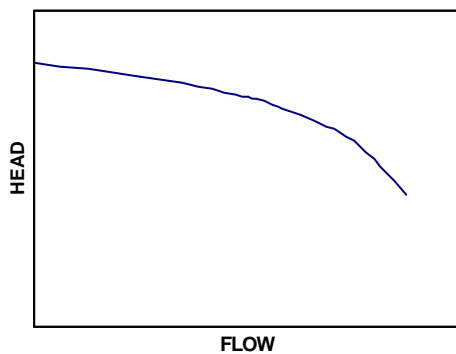


**Figure 2.5**  
System with low static head

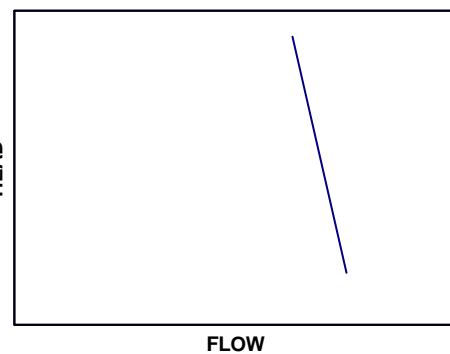
Static head is a characteristic of the specific installation and reducing this head where this is possible, generally helps both the cost of the installation and the cost of pumping the liquid. Friction head losses must be minimised to reduce pumping cost, but after eliminating unnecessary pipefittings and length, further reduction in friction head will require larger diameter pipe, which adds to installation cost.

### 2.3 PUMP CURVES

The performance of a pump can also be expressed graphically as head against flow rate. See Fig 2.6 for rotodynamic pumps and Fig 2.7 for positive displacement (PD) pumps.



**Figure 2.6**  
Rotodynamic Pump

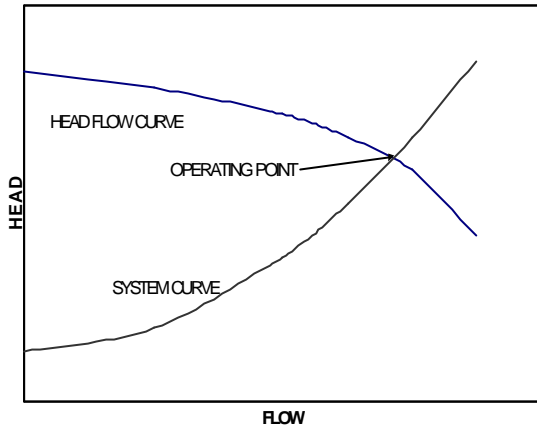


**Figure 2.7**  
Positive displacement Pump

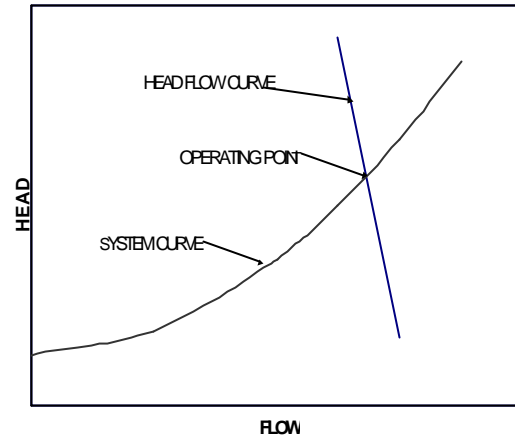
The Rotodynamic pump, (usually a centrifugal pump) has a curve where the head falls gradually with increasing flow, but for a PD pump, the flow is almost constant whatever the head. It is customary to draw the curve for PD pumps with the axes reversed (see Section 4), but to understand the interaction with the system, a common presentation is used here for the two pump types.

### 2.3 PUMP OPERATING POINT

When a pump is installed in a system the effect can be illustrated graphically by superimposing pump and system curves. The operating point will always be where the two curves intersect. (Fig 2.8 and Fig 2.9).



**Figure 2.8**  
**Rotodynamic pump with system curve**



**Figure 2.9**  
**P.D. pump with system curve**

If the actual system curve is different in reality to that calculated, the pump will operate at a flow and head different to that expected.

For a PD pump, if the system resistance increases, the pump will increase its discharge pressure and maintain a fairly constant flow rate, dependant on viscosity and pump type. Unsafe pressure levels can occur without relief valves.

For a rotodynamic pump, an increasing system resistance will reduce the flow, eventually to zero, but the maximum head is limited as shown. Even so, this condition is only acceptable for a short period without causing problems. An error in the system curve calculation is also likely to lead to a rotodynamic pump selection, which is less than optimum for the actual system head losses.

**Adding comfort margins to the calculated system curve to ensure that a sufficiently large pump is selected will generally result in installing an oversized pump, which will operate at an excessive flow rate or in a throttled condition, which increases energy usage and reduces pump life.**



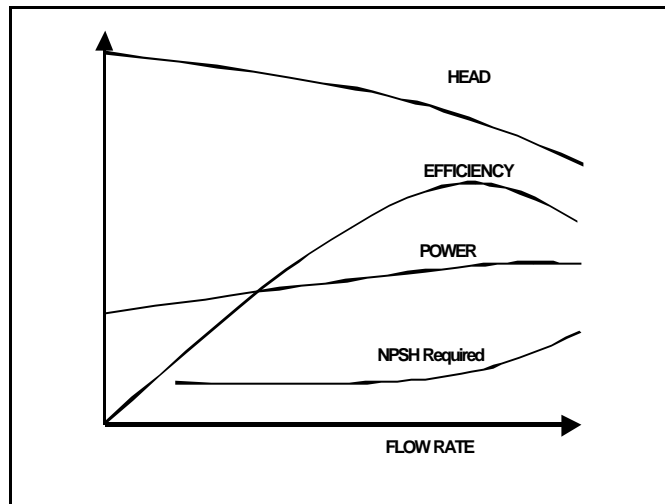
## SECTION 3 ROTODYNAMIC PUMPS

### 3.1 PUMP PRINCIPLES & PERFORMANCE CHARACTERISTICS

A rotodynamic or centrifugal pump is a dynamic device for increasing the pressure of liquid. In passing through the pump, the liquid receives energy from the rotating impeller. The liquid is accelerated circumferentially in the impeller, discharging into the casing at high velocity which is converted into pressure as effectively as possible.

Since the pump is a dynamic device, it is convenient to consider the head generated rather than the pressure. The pump generates the same head of liquid whatever the density of the liquid being pumped. The actual shapes of the hydraulic passages of the impeller and the casing are extremely important, in order to attain the highest efficiency possible.

The standard convention for rotodynamic pump is to draw the pump performance curves showing Flow on the horizontal axis and Head generated on the vertical axis. Efficiency, Power & NPSH Required (see Section 3.1.3 for explanation of NPSH), are also all conventionally shown on the vertical axis, plotted against Flow, as illustrated in Fig 3.1.



**Figure 3.1:**  
**Example of Pump performance curves**

#### 3.1.1 EFFECT OF SPEED VARIATION

As stated above, a centrifugal pump is a dynamic device with the head generated from a rotating impeller. There is therefore a relationship between impeller peripheral velocity and generated head. Peripheral velocity is directly related to shaft rotational speed, for a fixed impeller diameter and so varying the rotational speed has a direct effect on the performance of the pump. All the parameters shown in fig 3.1 will change if the speed is varied and it is important to have an appreciation of how these parameters vary in order to safely control a pump at different speeds. The equations relating rotodynamic pump performance parameters of flow, head and power absorbed, to speed are known as the Affinity Laws:

$$Q \propto N$$

$$H \propto N^2$$

$$P \propto N^3$$

Where:

$Q$  = Flow rate

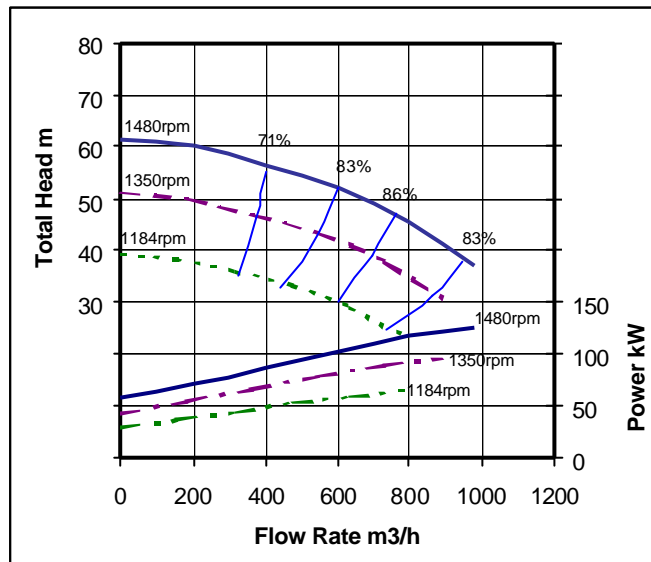
$H$  = Head

$P$  = Power absorbed

$N$  = Rotating speed

Efficiency is essentially independent of speed

The implication of the squared and cubic relationships of head and power absorbed, is that relatively small changes in speed give very significant changes in these parameters as shown in an example of a centrifugal pump in fig 3.2.



**Figure 3.2**

**Example of speed variation effecting rotodynamic pump performance.**

Points of equal efficiency on the curves for the 3 different speeds are joined to make the iso-efficiency lines, showing that efficiency remains constant over small changes of speed providing the pump continues to operate at the same position related to its best efficiency point (BEP).

The affinity laws give a good approximation of how pump performance curves change with speed but in order to obtain the actual performance of the pump in a system, the system curve also has to be taken into account, as will be explained later.

Magnetically driven pumps, with metallic containment shell, as well as the hydraulic power, which obeys the affinity laws, have a magnetic power absorbed, which follows a square law with speed.

The two types of power must therefore be calculated separately for a change of speed. In Appendix A1-1 this is explained further.

### 3.1.2 EFFECTS OF IMPELLER DIAMETER CHANGE

Changing the impeller diameter gives a proportional change in peripheral velocity, so it follows that there are equations, similar to the affinity laws, for the variation of performance with impeller diameter D:

$$Q \propto D$$

$$H \propto D^2$$

$$P \propto D^3$$

Efficiency varies when the diameter is changed within a particular casing. Note the difference in iso-efficiency lines in Figure 3.3 compared with Figure 3.2. The relationships shown here apply to the case for changing only the diameter of an impeller within a fixed casing geometry, which is a common practise for making small permanent adjustments to the performance of a centrifugal pump. Diameter changes are generally limited to reducing the diameter to about 75% of the maximum, i.e. a head reduction to about 50%. Beyond this, efficiency and NPSH are badly affected. However speed change can be used over a wider range without seriously reducing efficiency. For example reducing the speed by 50% typically results in a reduction of efficiency by 1 or 2 percentage points. The reason for the small loss of efficiency with the lower speed is that mechanical losses in seals and bearings, which generally represent <5% of total power, are proportional to speed, rather than speed cubed.

It should be noted that if the change in diameter is more than about 5%, the accuracy of the squared and cubic relationships can fall off and for precise calculations, the pump manufacturer's performance curves should be referred to.

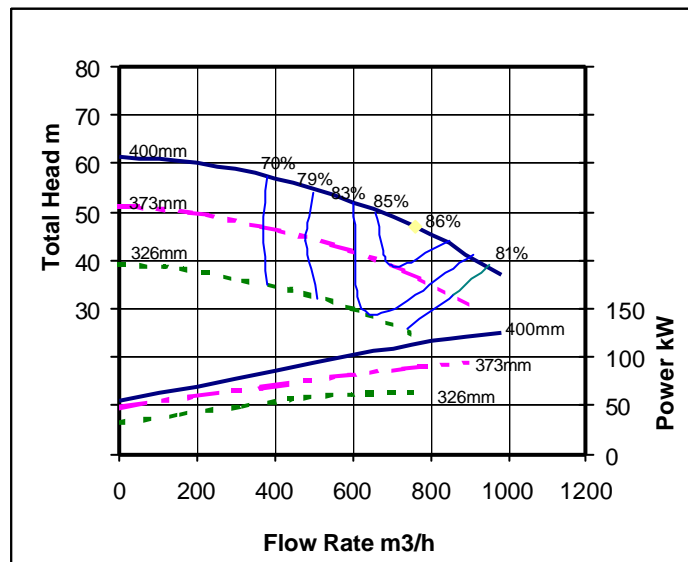


Figure 3.3

Example of impeller diameter reduction on rotodynamic pump performance.

The illustrated curves are typical of most rotodynamic pump types. Certain high flow, low head pumps have performance curve shapes somewhat different and have a reduced operating region of flows. This requires additional care in matching the pump to the system, when changing speed and diameter.

Magnetically driven pumps, may also need to be treated differently because a change of impeller diameter affects only the hydraulic power. Mechanical power loss in the drive is independent of diameter and so if the speed is unchanged the magnetic losses will not change.

See Appendix A1-2.

### 3.1.3 PUMP SUCTION PERFORMANCE (NPSH)

Liquid entering the impeller eye turns and is split into separate streams by the leading edges of the impeller vanes, an action which locally drops the pressure below that in the inlet pipe to the pump. If the incoming liquid is at a pressure with insufficient margin above its vapour pressure, then vapour cavities or bubbles appear along the impeller vanes just behind the inlet edges. This phenomenon is known as cavitation and has three undesirable effects:

- 1) The collapsing cavitation bubbles can erode the vane surface, especially when pumping water-based liquids.
- 2) Noise and vibration are increased, with possible shortened seal and bearing life.
- 3) The cavity areas will initially partially choke the impeller passages and reduce the pump performance. In extreme cases, total loss of pump developed head occurs.

The value, by which the pressure in the pump suction exceeds the liquid vapour pressure, is expressed as a head of liquid and referred to as Net Positive Suction Head Available – (NPSHA). This is a characteristic of the system design. The value of NPSH needed at the pump suction to prevent the pump from cavitating is known as NPSH Required – (NPSHR). This is a characteristic of the pump design.

The three undesirable effects of cavitation described above begin at different values of NPSHA and generally there will be cavitation erosion before there is a noticeable loss of pump head. However for a consistent approach, manufacturers and industry standards, usually define the onset of cavitation as the value of NPSHR when there is a head drop of 3% compared with the head with cavitation free performance. At this point cavitation is present and prolonged operation at this point will usually lead to damage. It is usual therefore to apply a margin by which NPSHA should exceed NPSHR.

As would be expected, the NPSHR increases as the flow through the pump increases, see fig 3.1. In addition, as flow increases in the suction pipework, friction losses also increase, giving a lower NPSHA at the pump suction, both of which give a greater chance that cavitation will occur. NPSHR also varies approximately with the square of speed in the same way as pump head and conversion of NPSHR from one speed to another can be made using the following equations.

$$Q \propto N$$

$$NPSHR \propto N^2$$

It should be noted however that at very low speeds there is a minimum NPSHR plateau, NPSHR does not tend to zero at zero speed

It is therefore essential to carefully consider NPSH in variable speed pumping.

### 3.2 METHODS OF VARYING PUMP PERFORMANCE

#### 3.2.1 THE NEED FOR PERFORMANCE VARIATION

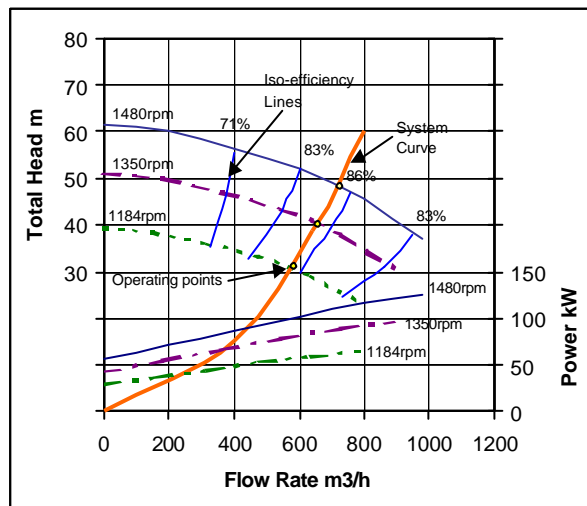
Many pumping systems require a variation of flow or pressure. To do so, either the system curve or the pump curve must be changed to get a different operating point. Where a single pump has been installed for a range of duties, it will have been sized to meet the greatest output demand, it will therefore usually be oversized, and will be operating inefficiently for other duties. There is therefore an opportunity to achieve an energy cost saving by using control methods which reduce the power to drive the pump during the periods of reduced demand. Not all control methods achieve this goal as explained in this section.

Varying pump performance by changing speed is explained first, it is the main focus of this guide, and in many cases is a cost effective approach with good pay back and even though the capital expenditure is relatively high, there can be savings on other equipment e.g. control valves. Other methods of control are then explained so that the most appropriate approach, to minimise life cycle cost, can be chosen. To make an effective evaluation of which control method to use, all of the operating duty points and their associated run time and energy consumption have to be identified, so that the total costs can be calculated and alternative methods compared.

Changing pump impeller diameter also effectively changes the duty point in a given system, (see Section 3.1.2), and at low cost, but this can be used only for permanent adjustment to the pump curve and is not discussed further as a control method.

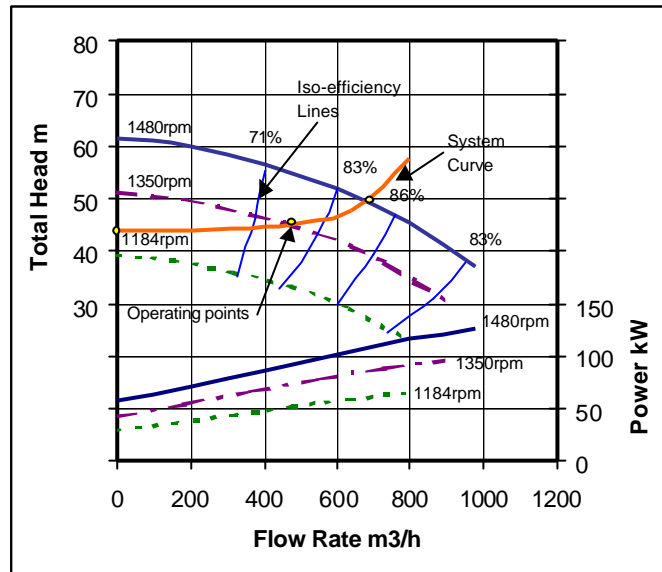
#### 3.2.2 PUMP CONTROL BY VARYING SPEED

To understand how speed variation changes the duty point, the pump and system curves are overlaid. Two systems are considered, one with only friction loss and another where static head is high in relation to friction head. It will be seen that the benefits are different. In Figure 3.4, reducing speed in the friction loss system moves the intersection point on the system curve along a line of constant efficiency. The operating point of the pump, relative to its best efficiency point, remains constant and the pump continues to operate in its ideal region. The affinity laws are obeyed which means that there is a substantial reduction in power absorbed accompanying the reduction in flow and head, making variable speed the ideal control method for systems with friction loss.



**Figure 3.4**  
**Example of the effect of pump speed change in a system with only friction loss**

In a system where static head is high, as illustrated in Figure 3.5, the operating point for the pump moves relative to the lines of constant pump efficiency when the speed is changed. The reduction in flow is no longer proportional to speed. A small turn down in speed could give a big reduction in flow rate and pump efficiency, which could result in the pump operating in a region where it could be damaged if it ran for an extended period of time even at the lower speed. At the lowest speed illustrated, (1184 rpm), the pump does not generate sufficient head to pump any liquid into the system, i.e. pump efficiency and flow rate are zero and with energy still being input to the liquid, the pump becomes a water heater and damaging temperatures can quickly be reached.



**Figure 3.5**

**Example of the effect of pump speed change with a system with high static head.**

The drop in pump efficiency during speed reduction in a system with static head, reduces the economic benefits of variable speed control. There may still be overall benefits but economics should be examined on a case -by -case basis.

Usually it is advantageous to select the pump such that the system curve intersects the full speed pump curve to the right of best efficiency, in order that the efficiency will first increase as the speed is reduced and then decrease. This can extend the useful range of variable speed operation in a system with static head. The pump manufacturer should be consulted on the safe operating range of the pump.

It is relevant to note that flow control by speed regulation is always more efficient than by control valve.

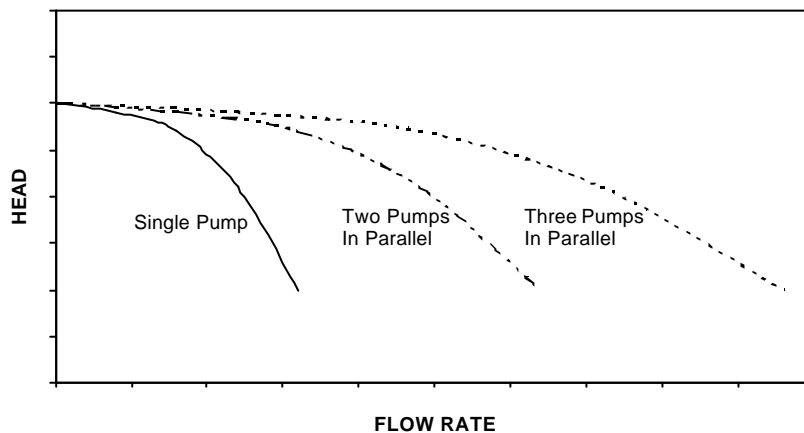
In addition to energy savings there could be other benefits of lower speed. The hydraulic forces on the impeller, created by the pressure profile inside the pump casing, reduce approximately with the square of speed. These forces, are carried by the pump bearings and so reducing speed increases bearing life. It can be shown that for a rotodynamic pump, bearing life is inversely proportional to the 7<sup>th</sup> power of speed. In addition, vibration and noise are reduced and seal life is increased providing the duty point remains within the allowable operating range.

The corollary to this is that small increases in the speed of a pump significantly increase power absorbed, shaft stress and bearing loads. It should be remembered that the pump and motor must be sized for the maximum speed at which the pump set will operate. At higher speed the noise and vibration from both pump and motor will increase, although for small increases the change will be small. If the liquid contains abrasive particles, increasing speed will give a corresponding increase in surface wear in the pump and pipework.

The affect on the mechanical seal of the change in seal chamber pressure, should be reviewed with the pump or seal manufacturer, if the speed increase is large. Conventional mechanical seals operate satisfactorily at very low speeds and generally there is no requirement for a minimum speed to be specified, however due to their method of operation, gas seals require a minimum peripheral speed of 5 m/s.

### 3.2.3 PUMPS IN PARALLEL SWITCHED TO MEET DEMAND

Another energy efficient method of flow control, particularly for systems where static head is a high proportion of the total, is to install two or more pumps to operate in parallel. Variation of flow rate is achieved by switching on and off additional pumps to meet demand. The combined pump curve is obtained by adding the flow rates at a specific head. The head/flow rate curves for two and three pumps are shown in Figure 3.6

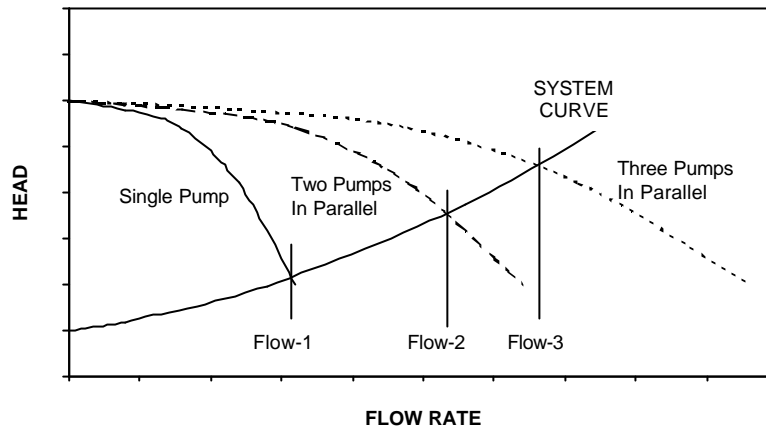


**Figure 3.6**  
**Typical head-flow curves for pumps in parallel**

The system curve is usually not affected by the number of pumps that are running. For a system with a combination of static and friction head loss, it can be seen, in Fig 3.7, that the operating point of the pumps on their performance curves moves to a higher head and hence lower flow rate per pump, as more pumps are started. It is also apparent that the flow rate with two pumps running is not double that of a single pump. If the system head were only static, then flow rate would be proportional to the number of pumps operating.

It is possible to run pumps of different sizes in parallel providing their closed valve heads are similar. By arranging different combinations of pumps running together, a larger number of different flow rates can be provided into the system.

Care must be taken when running pumps in parallel to ensure that the operating point of the pump is controlled within the region deemed as acceptable by the manufacturer. It can be seen from fig 3.7 that if 1 or 2 pumps are stopped then the remaining pump(s) would operate well out along the curve where NPSHR is higher and vibration level increased, giving an increased risk of operating problems.



**Figure 3.7**

**Typical Head-flow curves for pumps in parallel, with system curve illustrated.**

### 3.2.4 STOP/START CONTROL

In this method the flow is controlled by switching pumps on or off. It is necessary to have a storage capacity in the system e.g. a wet well, an elevated tank or an accumulator type pressure vessel. The storage can provide a steady flow to the system with an intermittent operating pump. When the pump runs, it does so at the chosen (presumably optimum) duty point and when it is off, there is no energy consumption. If intermittent flow, stop/start operation and the storage facility are acceptable, this is an effective approach to minimise energy consumption.

The stop/start operation causes additional loads on the power transmission components and increased heating in the motor. The frequency of the stop/start cycle should be within the motor design criteria and checked with the pump manufacturer.

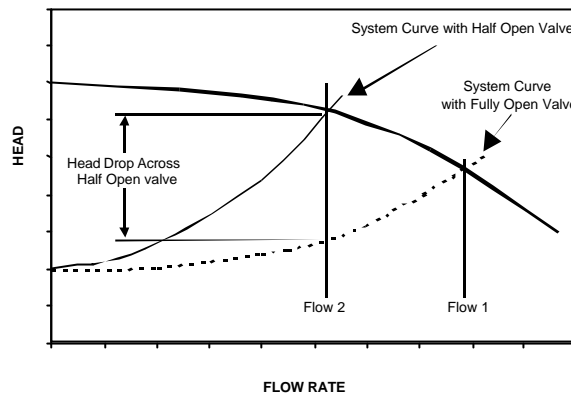
It may also be used to benefit from “off peak” energy tariffs by arranging the run times during the low tariff periods.

To minimise energy consumption with stop start control it is better to pump at as low flow rate as the process permits. This minimises friction losses in the pipe and an appropriately small pump can be installed. For example, pumping at half the flow rate for twice as long can reduce energy consumption to a quarter.



### 3.2.5 FLOW CONTROL VALVE

With this control method, the pump runs continuously and a valve in the pump discharge line is opened or closed to adjust the flow to the required value.



**Figure 3.8**  
Control of pump flow by changing system resistance using a valve.

To understand how the flow rate is controlled see Figure 3.8. With the valve fully open, the pump operates at “Flow 1”. When the valve is partially closed it introduces an additional friction loss in the system, which is proportional to flow squared. The new system curve cuts the pump curve at “Flow 2”, which is the new operating point. The head difference between the two curves is the pressure drop across the valve.

It is usual practice with valve control to have the valve 10% shut even at maximum flow. Energy is therefore wasted overcoming the resistance through the valve at all flow conditions.

There is some reduction in pump power absorbed at the lower flow rate (see Figure 3.1), but the flow multiplied by the head drop across the valve, is wasted energy. It should also be noted that, whilst the pump will accommodate changes in its operating point as far as it is able within its performance range, it can be forced to operate high on the curve where its efficiency is low, and where its reliability is impaired.

Maintenance cost of control valves can be high, particularly on corrosive and solids-containing liquids. So the lifetime cost could be unnecessarily high.

### 3.2.6 BY-PASS CONTROL

In this approach, the pump runs continuously at the maximum process demand duty, with a permanent by-pass line attached to the outlet. When a lower flow is required the surplus liquid is bypassed and returned to the supply source. An alternative configuration may have a tank supplying a varying process demand, which is kept full by a fixed duty pump running at the peak flow rate. Most of the time the tank overflows and recycles back to the pump suction. This is even less energy efficient than a control valve because there is no reduction in power consumption with reduced process demand.

The small by-pass line sometimes installed to prevent a pump running at zero flow is not a means of flow control, but required for the safe operation of the pump.

## **SECTION 4 POSITIVE DISPLACEMENT PUMPS**

### **4.1 PUMP PRINCIPLES, TYPES AND PERFORMANCE CHARACTERISTICS.**

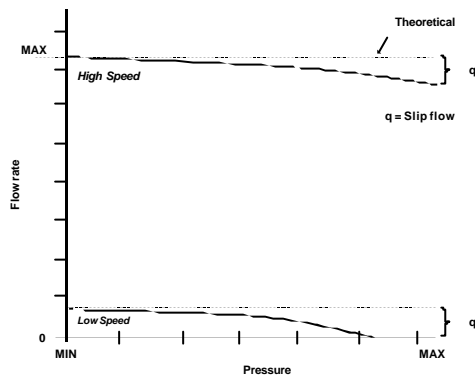
Positive Displacement Pumps can be classified into two main groups: Rotary and Reciprocating.

**Rotary pumps** (typical pressures up to 25 bar), transfer liquid from suction to discharge through the action of rotating screws, lobes, gears, valves, rollers etc, which operate inside a rigid casing. Rotary pumps do not require non-return valves on the inlet and outlet sides of the pump.

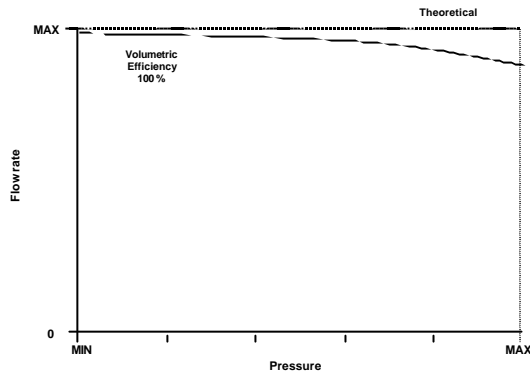
**Reciprocating pumps** (typical pressures up to 500 bar) discharge liquid by changing the internal volume of the pump. Reciprocating pumps incorporate both inlet and outlet non-return valves. These are generally integral with the pump body.

#### **Flow rate and Pressure**

The relationship between flow rate and pressure of the two types is shown in Figures 4.1 and 4.2.



**Figure 4.1**  
**Rotary**



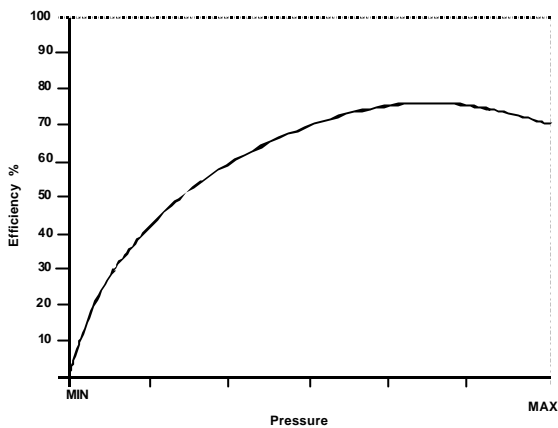
**Figure 4.2**  
**Reciprocating**

There is only a small fall off in flow rate with increasing pressure. This flow rate discrepancy is referred to as 'slip flow' for rotary pumps. Different types of PD pumps have different magnitudes of 'slip flow'

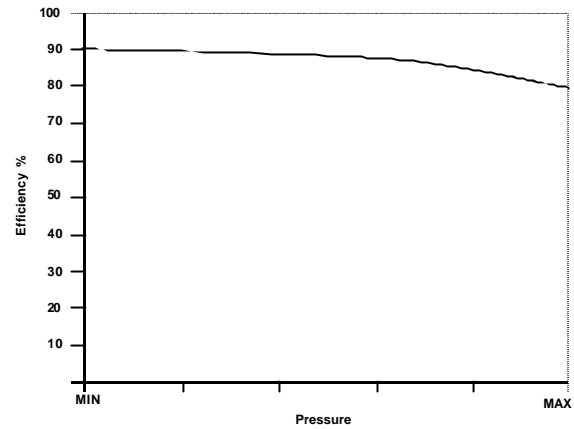
Discharge pressure will match the system's demand. Very high and dangerous pressures can be created by 'Dead Heading' i.e. operating the pump at zero flow. This condition is usually avoided, to comply with Statuary requirements, by fitting a pressure relief device before any isolating valve or potential blockage.

## EFFICIENCY

Typical pump efficiency curves are shown in figures 4.3 and 4.4 below.



**Figure 4.3**  
**Rotary Pump Efficiency**



**Figure 4.4**  
**Reciprocating Pump Efficiency**

## SUCTION PERFORMANCE ( NPIP)

In a similar way to that described in 3.1.3, a PD pump needs the incoming liquid to have a pressure margin above liquid vapour pressure to prevent cavitation in low-pressure areas in the suction passages of the pump. For a PD pump this required pressure is known as Net Positive Inlet Pressure (NPIP), sometimes referred to as Net Positive Suction Pressure. (NPSP)

## 4.2 METHODS OF VARYING PUMP PERFORMANCE

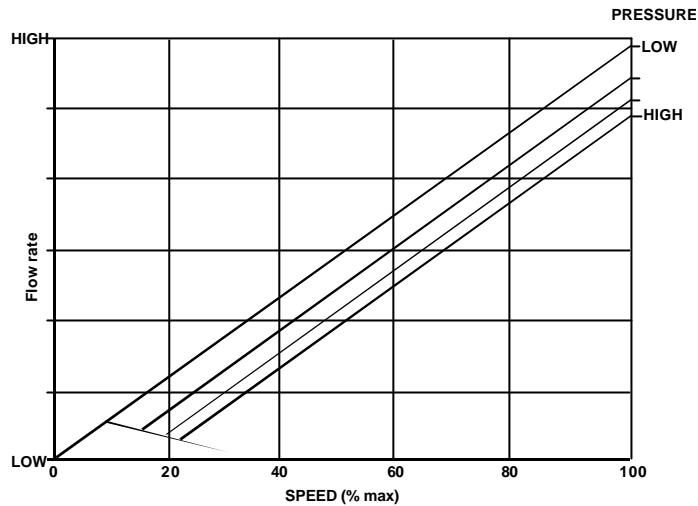
Unlike a rotodynamic pump, a PD pump cannot be controlled in flow by changing system resistance e.g. by closing a valve and so this simple but inefficient method is ineffective and potentially unsafe for PD pumps. Therefore, apart from changing stroke length on a limited number of reciprocating pumps, variable speed is the method generally used.

### 4.2.1 PUMP CONTROL BY VARYING SPEED

#### SPEED AND FLOW RATE

The flow rates of both types of PD pumps can be varied by changing the operating speed.

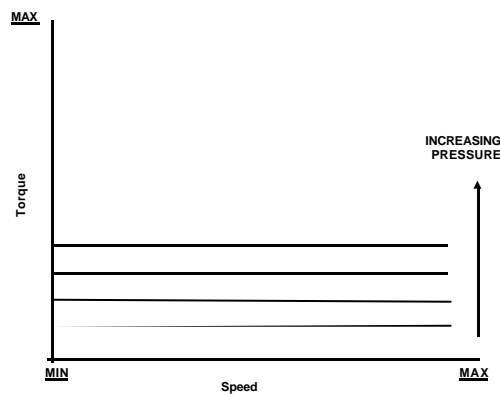
In general flow rates are directly proportional to speed and volumetric flow rate does not change significantly with pressure, below 200 bar. See figure 4.5 below.



**Figure 4.5**  
**Speed/Flow Rate**

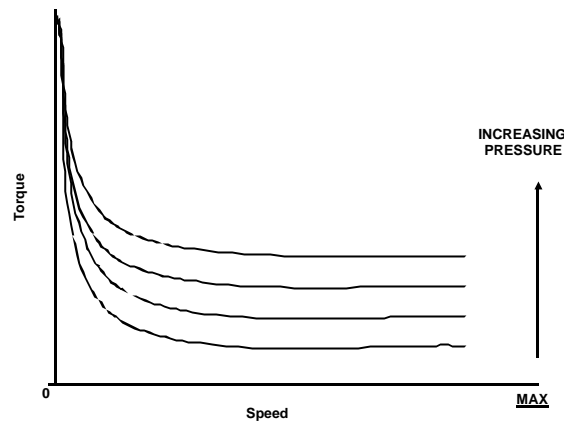
**Speed and Torque**

The torque required to drive a PD pump, in general, is directly proportional to the differential pressure across the pump and independent of speed (see Figure 4.6). Note that this is very different to the characteristic for a centrifugal pump where torque is proportional to speed squared.



**Figure 4.6**  
**Generalised PD Pump Speed Torque relation**

Some types of PD pump have a high starting torque; this can be a significant factor in sizing a drive. Some types of PD pumps (disc/diaphragm) have an increasing torque at the highest speeds. PD pumps, which depend on the pumped fluid for lubrication of their moving mechanical components, can show a torque characteristic which increases rapidly at low speed, see figure 4.7.

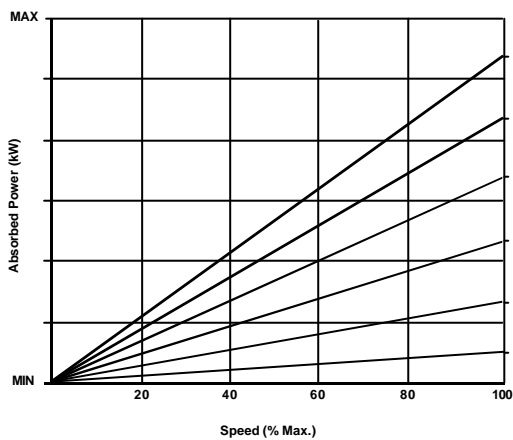


**Figure 4.7**  
**Progressive Cavity Pump Speed Torque Relation**

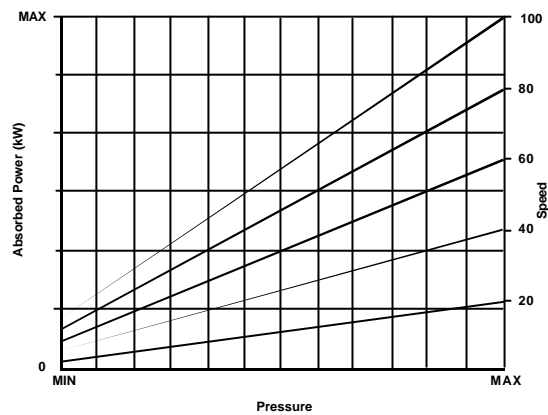
**Speed and Power Absorbed**

Typical power absorbed curves vary linearly with speed and pressure; examples are shown as figures 4.8 and 4.9. Flow control by varying speed is therefore the most efficient method, as it does not waste any of the input energy.

The liquid’s viscosity can have an effect on absorbed power and the liquid density effects the system, the pump manufacturer’s data should be consulted.



**Figure 4.8**  
**Rotary Pump Power Curve**



**Figure 4.9**  
**Reciprocating Pump Power Curve**

**Other Considerations of Variable Speed with PD Pumps**

The pressure can be maintained with reduced speed and flow, so large speed turndowns can be useful. It is important therefore to avoid a speed which is too low for stable and reproducible operation and to consider, at the system design stage, the constant torque characteristic and possible low speed torque effects, because of its demand on electronic variable speed drives. In particular low speed running at constant torque requires that motor cooling is carefully studied.

Running some types of pump too slowly can have detrimental effects on wear rates when handling slurries containing settling solids. PD pumps with directly driven lubrication systems may also have limitations on minimum speeds.

PD pumps are not inherently high in noise and vibration and any perceived noise from a pump unit can usually be attributed to either the drive, or to hydraulic noise in the associated pipes and valves. Some PD pumps do contain out of balance moving components, but increasing speed within practical limits does not normally produce abnormal vibration.

#### **4.2.2 FLOW CONTROL USING PUMPS IN PARALLEL**

PD pumps can generally be run in parallel without problems. This gives increased flow rates at the pressure rating of a single pump. The principle considerations are, the correct design of inlet and outlet pipe work to avoid problems of NPIP in the inlet, over pressure in the discharge pipe work, and back-flow through a stationary pump.

Different PD pumps can be operated in parallel; they do not have to be 'matched'.

With reciprocating pumps, synchronising the strokes in order to minimise pressure pulsation is a consideration.

#### **4.2.3 PUMPS IN SERIES (ROTARY)**

Rotary PD pumps can be run in series, this gives increased pressure capability at the flow rate of a single pump. Careful design of the control logic and overpressure prevention/relief are important.

Matching the speed of the two pumps is important in order to avoid over or under feeding of the secondary pump. Over feeding the secondary pump will cause overpressure of the primary pump.

Under feeding the secondary pump will cause cavitation and NPIP problems.

The use of one motor to drive both pumps ensures synchronised starting of the two pumps and speed variations, due to the electrical supply or motor loading, are automatically compensated. Some types of PD pumps (e.g. progressive cavity) tend to balance out any small mismatches in flow rates and this minimises operational problems.

#### **4.2.4 FLOW CONTROL VALVE**

This is not an acceptable technique. Throttling a PD pump will change pump pressure but will not change the flow rate and can lead to excessive pressures.

#### **4.2.5 BY-PASS CONTROL**

Control is by either 'modulating' or on/off control of the bypass flow, but is not commonly used because of the energy wasted and wear on control valves in the bypass circuit.

#### **4.2.6 LOAD/UNLOAD CONTROL**

Unlike a rotodynamic pump this can be an effective way of controlling a PD pump discharge rate.

Load/unload control is similar to bypass control and energy wastage is relatively low, however, wear of the load/unload valve is a problem.

## **SECTION 5 MOTORS**

Whilst there are many types of prime movers available (such as diesel engines, steam turbines, dc motors, permanent magnet motors, synchronous reluctance motors and wound rotor motors) the majority of pumps are driven by an ac induction motor. Although this document is principally about pumps and Variable Speed Drives it should be mentioned that, on a typical industrial site, motor driven equipment accounts for approximately two thirds of electricity costs and improvements in motor efficiency can offer major energy savings. The principles outlined will apply to all motors on a given site, not just those used as pump drivers.

### **5.1 MOTOR PRINCIPLES**

The speed of an induction motor is normally fixed because the supply frequency is fixed, as is the number of poles in the motor. The speed ( ignoring slip ) is calculated from the formula :

$$\text{SPEED (R/MIN)} = \frac{120 \times \text{SUPPLY FREQUENCY (HZ)}}{\text{number of poles}}$$

i.e.: - a 2 pole motor on 50 Hz supply has a speed of  $\frac{120 \times 50}{2} = 3000$  rpm

Equally a 2 pole motor on a 60 Hz supply has a speed of 3600 r/min

Therefore, by varying the frequency the speed can also be varied.

### **5.2 MULTI-SPEED MOTORS**

Varying the frequency can give a step-less change of speed, but if a small number of predetermined speeds are acceptable, a multi-speed motor is an effective solution.

Two, three or four fixed speeds can be achieved by special windings within the same stator or frame and a dedicated controller.

### **5.3 ENERGY EFFICIENCY**

The average electric motor will consume its capital cost in energy in less than 2 months, typically a motor, costing £500 will consume over £50000 in its lifetime. Therefore a single percentage point increase in efficiency will save lifetime energy cost generally equivalent to the purchase price of the motor. This illustrates the importance of giving close attention to efficiency criteria.

The calculation for the energy cost per annum of any electric motor is:

$$\frac{\text{Hours used per year} \times \text{kWh tariff} \times \text{operating point kW}}{\text{Efficiency at operating point}}$$

Typically for a pumping system: -

Design duty point	80 kW	Installed motor rating	90 kW
Operating point	67.5 kW	Operating	6000 hrs/yr
Motor efficiency	95.0%	Tariff	4.5 p/kWh

$$\text{Energy cost} = \frac{6000 \times \text{£}0.045 \times 67.5}{0.95}$$

$$= \text{£ } 19,184 \text{ per year}$$

Using this formula, comparisons can be made between different types of motor. Based on a typical fourteen-year life of an electric motor, lifetime cost savings for high efficiency motors are in the order of 3-4 times the purchase cost.

Efficiency depends not only on motor design, but also on the types and quantity of active materials used. The efficiency can therefore vary considerably from manufacturer to manufacturer.

Manufacturers have focused on the following key factors to improve the efficiency of a motor:

- Electromagnetic design – Making the best use of copper by winding techniques and lamination design.
- Magnetic steel – Utilising a low loss, high permeability steel.
- Thermal design – Ensuring optimum fit between stator, frame and laminations.
- Aerodynamics – Using a more efficient cooling system by change of fan and/or fan cover design.
- Manufacturing quality – Improving assembly techniques.

By adopting these techniques, manufacturers have made efficiency improvements in the range of 3%, on motors up to 400kW. The percentage gains on the lower kW output motors could be greater than 3%, the gains on the higher kW output motors will not be as great.

There are several international standards for measuring the efficiency of a motor. European (IEC 34) and North American (IEEE 112) standards vary and will inevitably produce differing results. In comparing any manufacturers' data, the supply input and test method utilised must be common to each set of data.

#### 5.4 EFFICIENCY LABELLING

With the backing of the European Commission, manufacturers representing 80% of the European production of standard motors have agreed to establish three efficiency bands or classes for their 3-phase TEFV (totally enclosed fan ventilated), 2 and 4 pole, cage induction motors in the range 1.1-90 kW. Most industrial equipment manufacturers and users are therefore affected by this scheme which covers the vast majority of 3 phase motors purchased.

The agreement also commits manufacturers, who sign up to the scheme, to a 50% reduction in the supply of motors in the lowest efficiency class, 'Eff 3', by the end of 2003.

The efficiency class of each motor, within the scope of the scheme, will be identified in catalogues and on nameplates as a minimum. Some manufacturers will go further and include stickers on the motors for greater visibility. This will assist equipment manufacturers to demonstrate their more energy efficient equipment to their end-user customers. These labels may, in some cases, be incorporated into other stickers or branding used by the manufacturer to identify his product range. Figure 5.1<sup>1</sup> below illustrates the minimum basic labels.

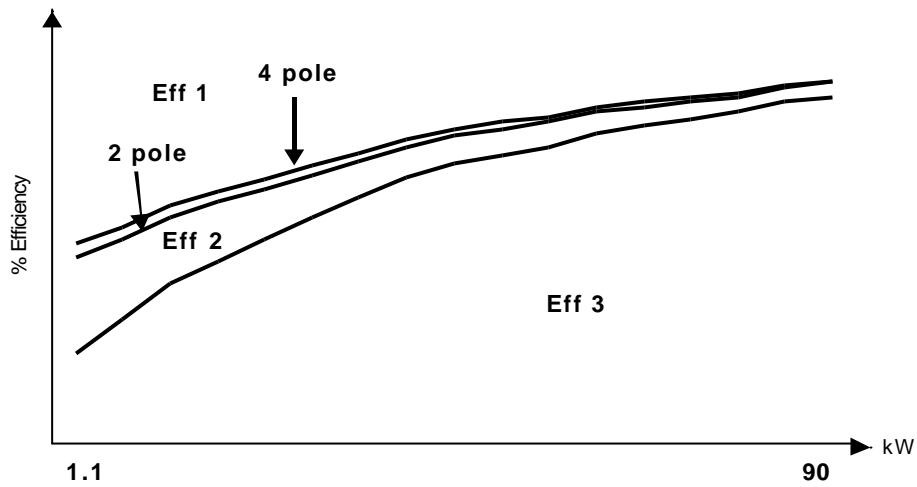


**Basic efficiency classification labels**

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<sup>1</sup> Trade Marks of Gemelec





**Figure 5.2**  
**Graph of Efficiency Classification Bands**

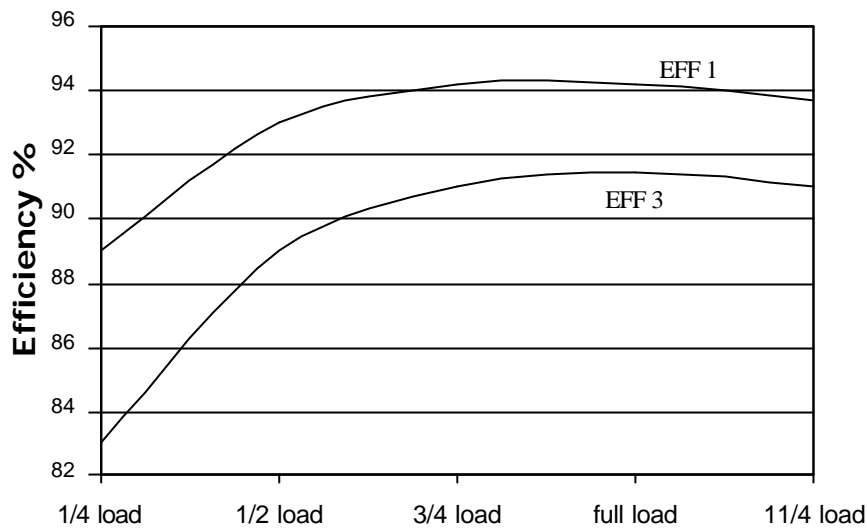
Motor efficiency classification bands are illustrated in Figure 5.2. It is apparent that the spread of Eff 2 motor efficiencies is very wide, (up to 7.6%) in smaller sizes and becomes much narrower (1.1%) in larger sizes. The convergence in larger sizes is realistic since most motors in larger sizes have similar numerical efficiencies, but small differences are very significant in terms of lifetime energy consumption and cost for larger motors.

## 5.5 OTHER ENERGY SAVING OPPORTUNITIES

### 5.5.1 MOTOR SIZING

Electric motors are designed to operate at full rated output, at rated voltage, twenty-four hours per day, three hundred and sixty five days per year. However, it is estimated that only 20% of machines in operation are running at their full rated output. The practice of utilising a 10% or perhaps 15% margin can often lead to the selection of a higher power rating and, in some cases an increase in the physical size, and therefore cost of the machine.

The loading of the motor affects its efficiency; Fig 5.3 shows a typical comparison of efficiency between a standard and a high efficiency motor. In most cases the motor will be operating below its rated output. As can be seen in the diagram, the difference in efficiency at full load between the standard and high efficiency motor may be as small as 2 – 3%, but at half load the difference is considerably greater.



**Figure 5.3**  
**Relative Efficiency Levels of Motors**

### 5.5.2 SWITCH IT OFF!

The first rule of energy savings is ‘If it isn’t being used, switch it off’. This is a low cost maxim, which has great effect, but is not frequently enough applied. Put simply, the operator does not feel the pain of the energy expenditure.

### 5.5.3 A MOTOR MANAGEMENT POLICY

When a motor is rewound, its efficiency is reduced, and the modest cost saving of the rewind compared with a new machine, is very soon lost against the additional energy losses. Institute a motor management policy to provide a structured approach to your replace/repair decisions. Rewinds should be in accordance with the best practices detailed by the Association of Electrical and Mechanical Trades (AEMT<sup>2</sup>). Replacements should be high efficiency, designated Eff1, and correctly sized for the application.

<sup>2</sup> AEMT Best Practice Guide

#### **5.5.4 SHAFT ALIGNMENT**

Misalignment of motor couplings is also surprisingly wasteful. An 0.6 mm angular offset in a pin coupling can result in as much as 8% power loss and eventual coupling failure with attendant production downtime. Check and realign motor drive couplings, starting with the largest motors.

#### **5.5.5 PULLEY SIZING**

Significant energy savings can be often be made simply by changing pulley sizes, to ensure a fan or pump runs at a more appropriate duty point. This doesn't provide the flexibility of variable speed control but costs very little and can probably be done within the maintenance budget and doesn't require capital approval.

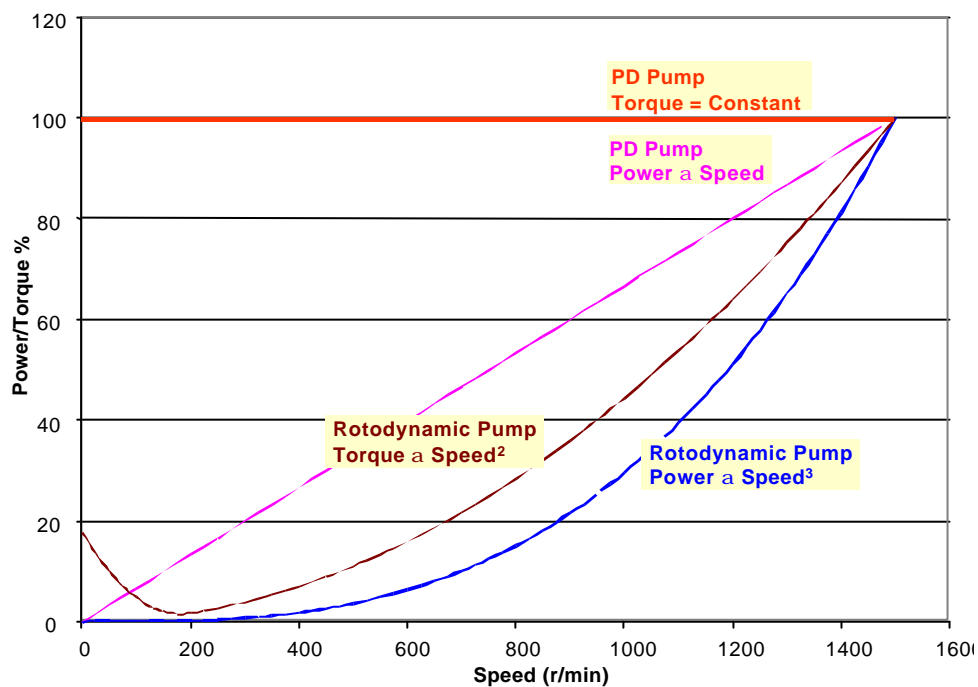
## **SECTION 6 VARIABLE SPEED DRIVES**

### **6.1 VARIABLE FREQUENCY DRIVE PRINCIPLES**

As seen earlier in section 5.1, a motor is capable of operating over a range of speeds if correctly fed at a varying frequency.

In section 3.1.1 we have seen that a rotodynamic pump performance curves show a power demand that follows the affinity laws, and therefore torque is proportional to (speed)<sup>2</sup>. See also Fig 6.1. This means that in principal a rotodynamic pump (without influence from the system curve), when slowed by 10% will demand only around 70% of the energy at full speed.

For a great majority of Positive displacement pumps, torque remains constant over the operating speed range. This is significant in the selection of the drive system, and in determining motor derating



**Figure 6.1**  
**Power and torque vs. speed**

### **6.2 THE FREQUENCY CONVERTER**

The most commonly used type of electronic variable speed drive is a frequency converter used in conjunction with an induction motor.

The frequency converter may be referred to by several terms and abbreviations, including an inverter (which is only part of the converter system), or as a VVVF (variable voltage, variable frequency drive) also VFD (variable frequency drive).

Irrespective of type, a frequency converter will consist of four basic parts, and the combination of these parts will affect the final performance of the system. Parts described below in 6.2.1 to 6.2.4 and Figure 6.2

In addition to the electronics described here the drive system will require conventional switching components in the supply and safety circuitry.

### 6.2.1 RECTIFIER

A frequency converter will operate by rectifying the incoming AC supply to a DC level. The type of rectifier can vary depending on the type of performance required from the drive.

The rectifier design will essentially control the harmonic content of the rectifier current, as the rectifier may not draw current for the full cycle of the incoming supply. It will also control the direction of power flow.

### 6.2.2 INTERMEDIATE CIRCUIT

Having rectified the incoming AC supply, the resultant will be an uneven rectified DC. This is smoothed in the intermediate circuit, normally by a combination of inductors and capacitors.

Over 98% of drives currently in the marketplace use a fixed voltage DC link.

### 6.2.3 INVERTER

The inverter stage converts the rectified and smoothed DC back into a variable AC voltage and frequency.

This is normally done with a semiconductor switch. The most common switches in low voltage systems are currently IGBTs – Insulated Gate Bipolar Transistors. To complete the circuit when one semiconductor is switched on, each switch is bridged in the reverse polarity by a “flywheel diode”.

### 6.2.4 CONTROL UNIT

The control unit gives and receives signals to the rectifier, the intermediate circuit and the inverter to achieve the correct operation of the equipment.

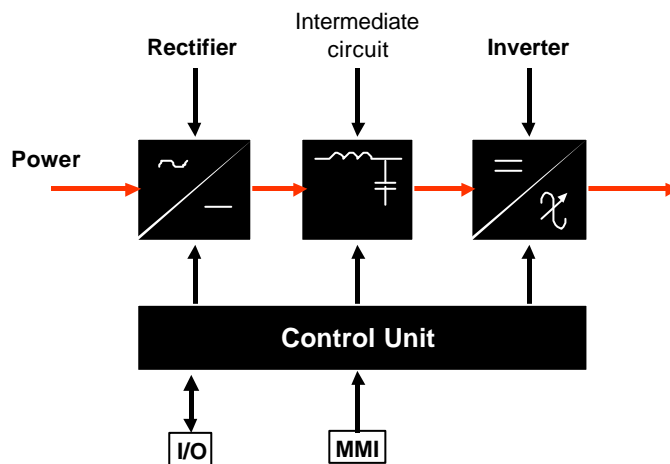


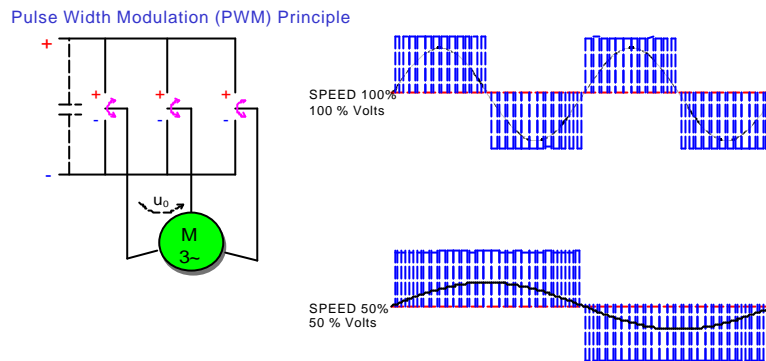
Figure 6.2

Basic Elements of frequency converter

### 6.3 PULSE WIDTH MODULATION

For commercial purposes the inverter control may be described differently, however, there is one fundamental technique that is used – Pulse Width Modulation (PWM)

The principle of PWM is to generate a pulse train by switching between the positive and the negative legs of the DC link, to simulate the required voltage and frequency as shown in fig 6.3 below.



**Figure 6.3**

#### **PWM Waveform Generation**

When a frequency converter is connected to a motor, it provides a voltage and frequency; the current is controlled solely by the motor internal impedances.

One major benefit of this is that there is no starting inrush current from the supply, compared to a direct on line system, allowing soft start both electrically and mechanically.

### 6.4 INTEGRATED VARIABLE SPEED MOTORS

A recent trend amongst many of the motor manufacturers has been to develop a motor with an integral inverter. These packages are currently available to approximately 7.5kW. The major advantages to the pump designer are include the reduced control panel space, the simplified installation, the straightforward electromagnetic compatibility compliance, ensured matching of motor and drive leading to greater life cycle cost savings.

## **SECTION 7 SELECTING A VARIABLE SPEED DRIVE FOR A NEW INSTALLATION**

### **7.1 SIZING & SELECTION**

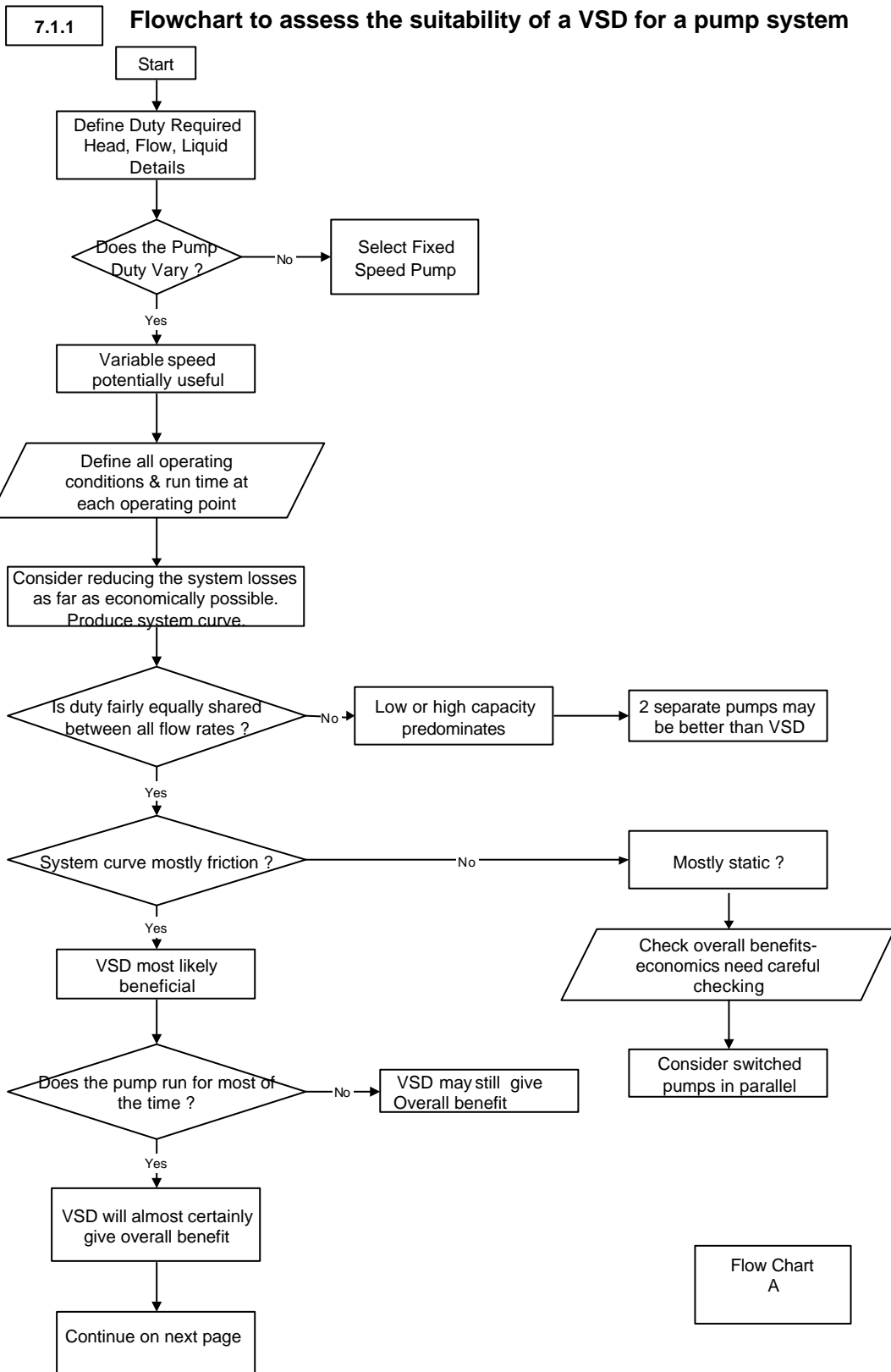
When the pump maximum duty is known, the peak power and speed for the drive will become clear. It is essential to commence the sizing exercise with the hydraulic system, and to work systematically to select the pump, motor and drive.

It would be normal to add a tolerance equal to the potential fall off in efficiency over the maintenance life of the system. It is recommended when selecting a Rotodynamic pump that the maximum flowrate is to right hand side of the best efficiency point.

Some operating profiles will be best satisfied by sharing the duty between multiple Variable Speed Driven Pumps.

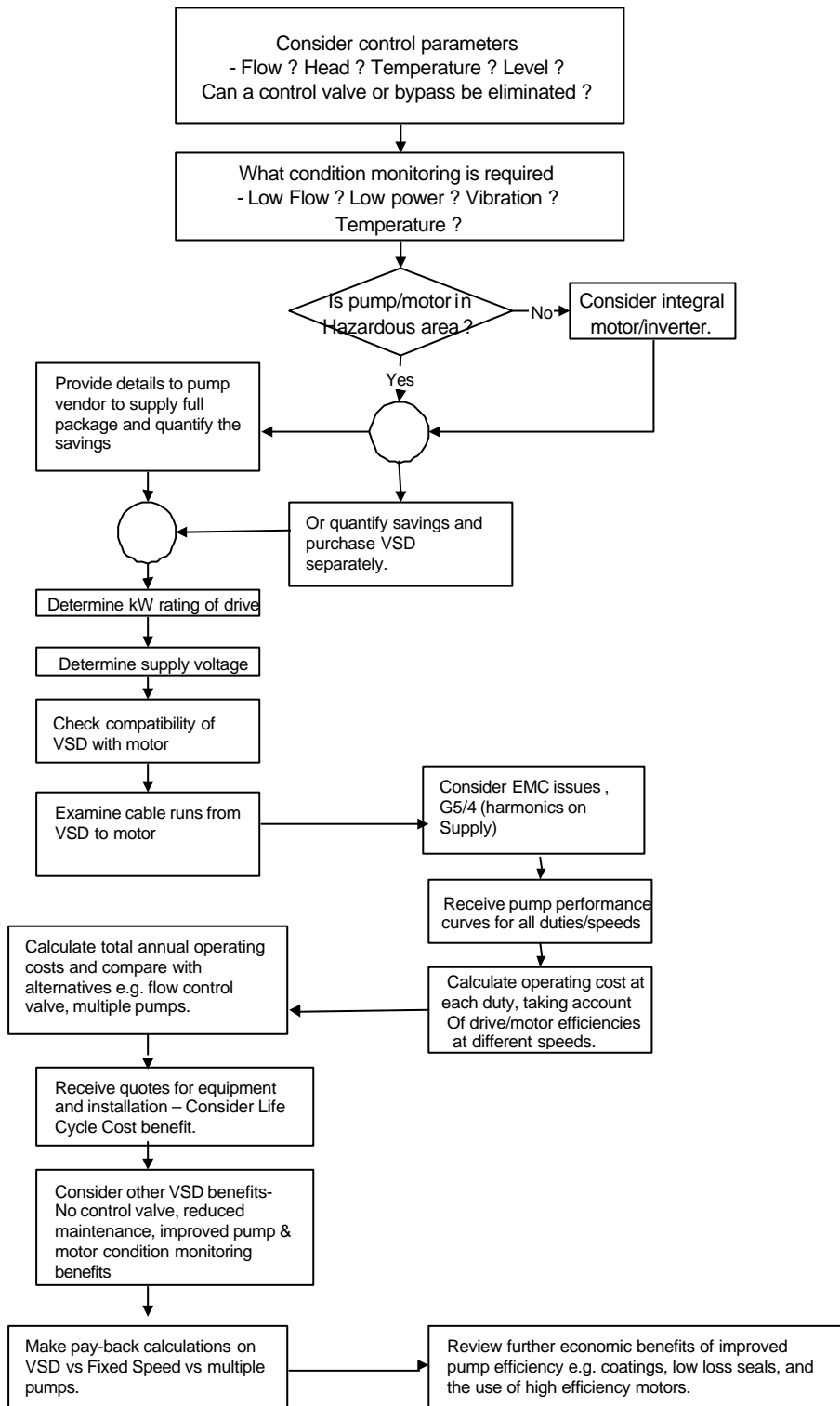
The following selection flow chart (A) is a step-by-step approach to see if a Variable Speed Driven pump will benefit the system and explain the decisions required.

Throughout this process we recommend consultation with the suppliers whose expertise should assist your selection.





**Continuation of Flowchart (A) to assess the suitability of a VSD for a pump system**



## 7.2 CONTROL

A pump with variable speed capability will need to be controlled to unlock all the benefits available from VS operation.

The effect of varying speed with a centrifugal pump is to vary both head and flow. Variation of speed with a positive displacement pump will vary only the flow rate.

<u>Process</u>	<u>Controlled Parameter</u>	<u>External influence</u>
Heating system	Temperature	Ambient
Tank filling	Level	Outflow
Pipeline	Flow	Level
Distribution	Pressure	Draw-off

**Table 7.1**

### Examples of control parameter

#### 7.2.1 CONTROL BY FIXING PRESSURE BUT VARYING FLOW

The most common form of control is by use of a discharge pressure sensor which sends a signal to the VFD which in turn varies the speed allowing the pump to increase or decrease the flow required by the system.

This form of control is common in water supply schemes where a constant pressure is required but water is required at different flows dependant on the number of users at any given time. Capacity changes at constant pressure are also common on centralised cooling and distribution systems and in irrigation where a varying number of spray heads or irrigation sections are involved.

#### 7.2.2 HEATING SYSTEM CONTROL

In heating and cooling systems there is a requirement for flow to vary based on temperature.

In this instance the VFD is controlled by a temperature sensor, which allows the flow of hot, or cold liquid in the system to increase or decrease based on the actual temperature required by the process.

This is similar in operation to pressure control, where the flow is also the variable entity, but a constant temperature requirement from a temperature sensor replaces that from a pressure sensor.

### **7.2.3 CONTROL BY FIXING FLOW BUT VARYING PRESSURE:**

In irrigation and water supply systems constant flow is often required, even though the water levels both upstream and downstream of the pumping station vary.

Also many cooling, chiller, spraying and washing applications require a specific volume of water to be supplied even if the suction and delivery conditions vary. Typically suction conditions vary when the height of a suction reservoir or tank drops and delivery pressure can change if filters blind or if system resistance increases occur through blockages etc.

The VSD system is usually the optimum choice to keep constant the flow rate in the system using a control signal from a flowmeter, which can be, installed in the suction, but more commonly the discharge line.

### **7.2.4 IMPLEMENTATION**

In many cases there will be an external control system, such as a PLC or PC, which will provide the start/stop control and an analogue speed reference to the drive or will pass this information to the drive by a serial communications link. In other cases the drive may have adequate on board intelligence.

All modern drive systems rely on microprocessor control, and this allows the manufacturer to integrate the basic signal processing functions into the drive. In some instances manufacturers have installed bespoke control software to the drive to allow specific requirements to be met.

As every case has its own specific requirements, it is important that the control requirements are understood in order to achieve the optimum system performance.

### **7.2.5 SOFT STARTING AND STOPPING**

When a induction motor is started direct on line, it will generate a high level of torque, which will cause a very fast breakaway, and it will then accelerate up to speed in an uncontrolled fashion.

In this case the network has to supply a large inrush current, a very high initial level to establish flux in the motor, followed by a high level which decreases as the drive accelerates.

The effect on the pump is to place mechanical stresses on the rotating components, followed by stresses in the hydraulic system, which may include a high initial flow rate causing a vacuum to be drawn on the suction side, or surge on the discharge, and possible NPSH problems.

Equally when stopping the rate of deceleration is totally uncontrolled, which can lead to further mechanical stresses, and surges in the hydraulic circuit. This can lead to requirements for additional inertia added to a pump, generally in the form of a flywheel, or to surge control vessels in the hydraulic system.

The use of electronic starting systems allows smooth acceleration and deceleration of a drive system.

Electronic soft starters will reduce the voltage at the motor terminals in a controlled manner, but are generally short time rated devices, while a frequency converter is usually continuously rated and so can be used to give very controlled rates of change.

The only drawback with either electronic scheme is that generally the equipment must be connected to the network, and therefore problems of uncontrolled deceleration could arise in a power failure.

## **SECTION 8 RETROFITTING A VARIABLE SPEED DRIVE TO EXISTING EQUIPMENT**

### **8.1 JUSTIFICATION**

There are approximately 20 times more pumps in service in existing installations than are supplied new every year.

It is common for fixed speed pumps to be oversized; most system designers allow a contingency on the system head required. It follows therefore that retrofitting variable speed drives could match pumps to actual system requirements more accurately and save considerable amounts of energy.

Also many existing systems use control valves and bypasses, all of which absorb energy not required to satisfy system demands.

It is therefore apparent that a major opportunity exists for modifying installed systems to make them more energy efficient. The fitting of variable speed drives and the removal of control valves and bypasses will save energy and often the payback for the modifications is short. Just consider a speed reduction of 10% with a rotodynamic pump will save approximately 30% of the electrical energy absorbed and 75% of pump systems are oversized, many by between 10 and 20%

Pump Manufacturers should be contacted to ensure pumps can be run at slower speeds with no detrimental effects and the motor manufacturers to ensure motors are suitable for use with variable speed drives. There may also be other alternatives; for example if a pump is oversized but operates at a single duty, the impeller diameter can be permanently reduced, which achieves the same energy reduction as a VSD but at a small cost.

### **8.2 MOTOR DERATING**

When considering adding a variable speed drive to an existing motor, care should be taken to match the electrical characteristics of the motor and frequency converter; otherwise the risk of premature failure is introduced into the system.

Early frequency converters produced outputs that had a very high harmonic content in the waveform. This resulted in substantial additional heating of motor windings, and therefore motors were de-rated for inverter use. Modern inverter outputs, cause relatively small levels of harmonic current distortion in the motor windings, and therefore little de-rating is normally required. Whilst this de-rating will be minimal it will vary from one motor design to another. It is also dependent on the type of inverter used.

The following parameters vary from one converter design to another and will affect the drive system performance:

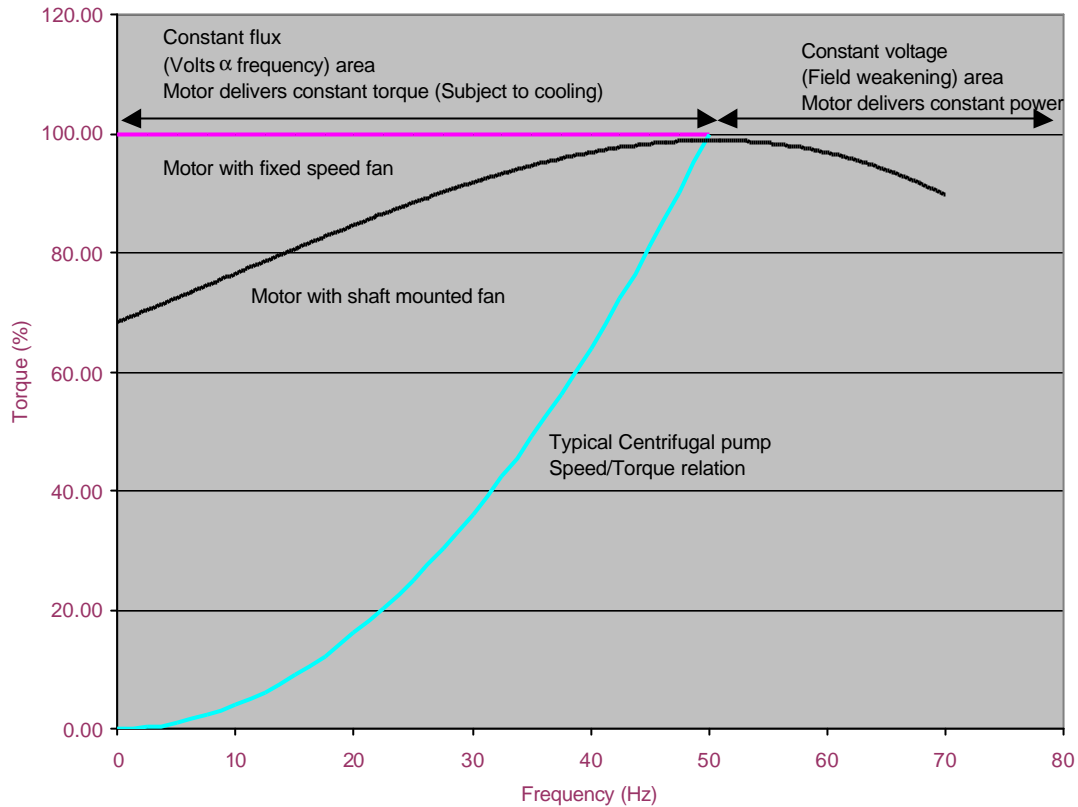
- Total harmonic distortion
- Peak voltage
- Maximum rate of change of voltage
- Switching frequency
- Cable length between the motor and inverter

Motor de-rating may also be required to compensate for the reduced cooling at lower speeds when the motor shaft mounted fan is not generating the airflow achieved at normal synchronous speed. The motor manufacturer's expertise must be used to determine how much de-rating is required and their recommendations applied. There are no definitive rules that can be applied to all motors.

## VARIABLE SPEED DRIVEN PUMPS-BEST PRACTICE GUIDE.

In general, little or no de-rating is required with most rotodynamic pump drives, however PD pumps with a constant torque characteristic are more likely to require de-rating of the motor for reduced speed operation.

Typically motors with a fixed speed-cooling fan can deliver a constant torque continuously over a wider speed range.



**Figure 8.1**

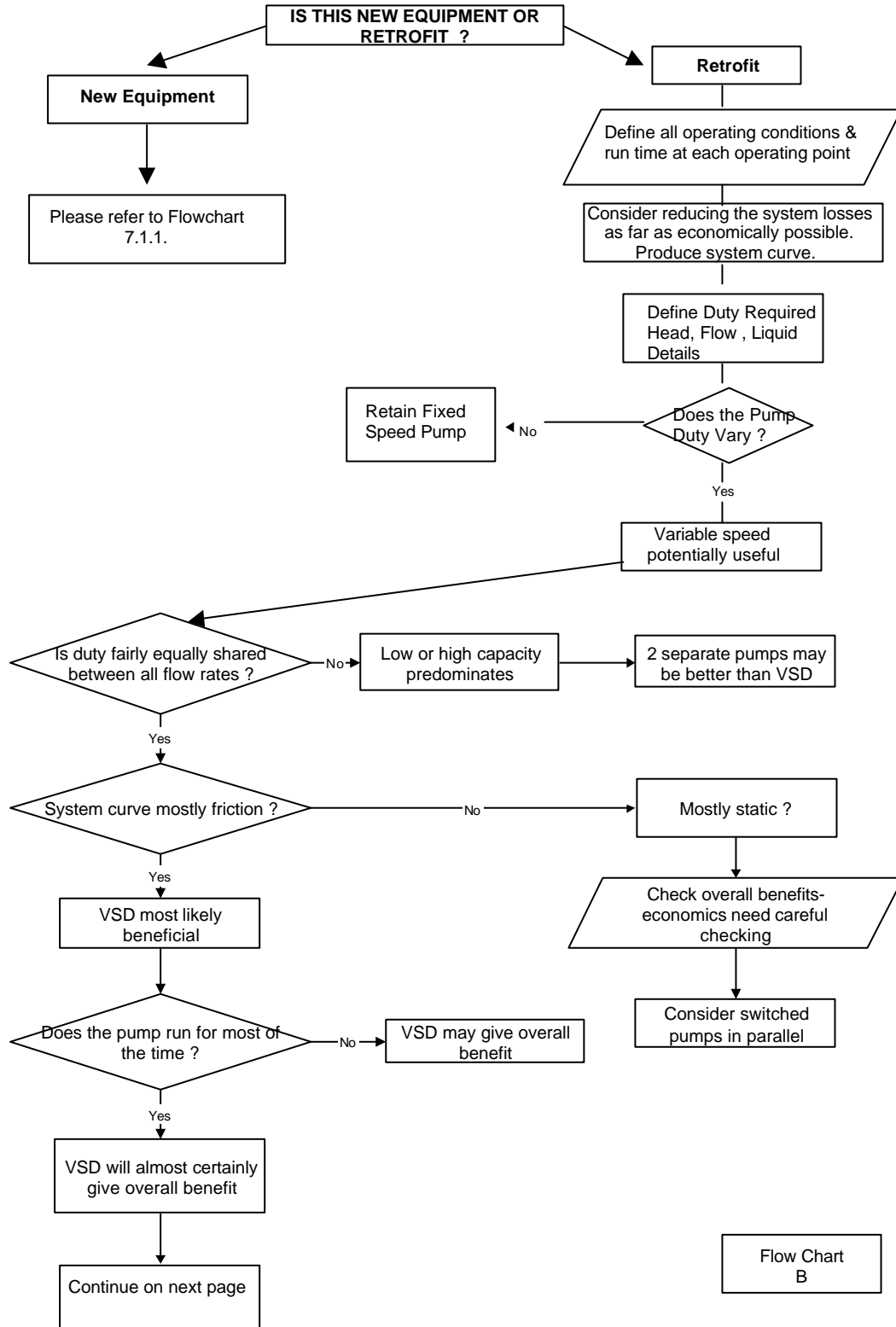
### Typical Isothermal Loadability Curves

Figure 8.1 shows the effect of varying the frequency (speed) of a typical motor, with a power rating based on Class B ( $80^{\circ}$  K temperature rise).

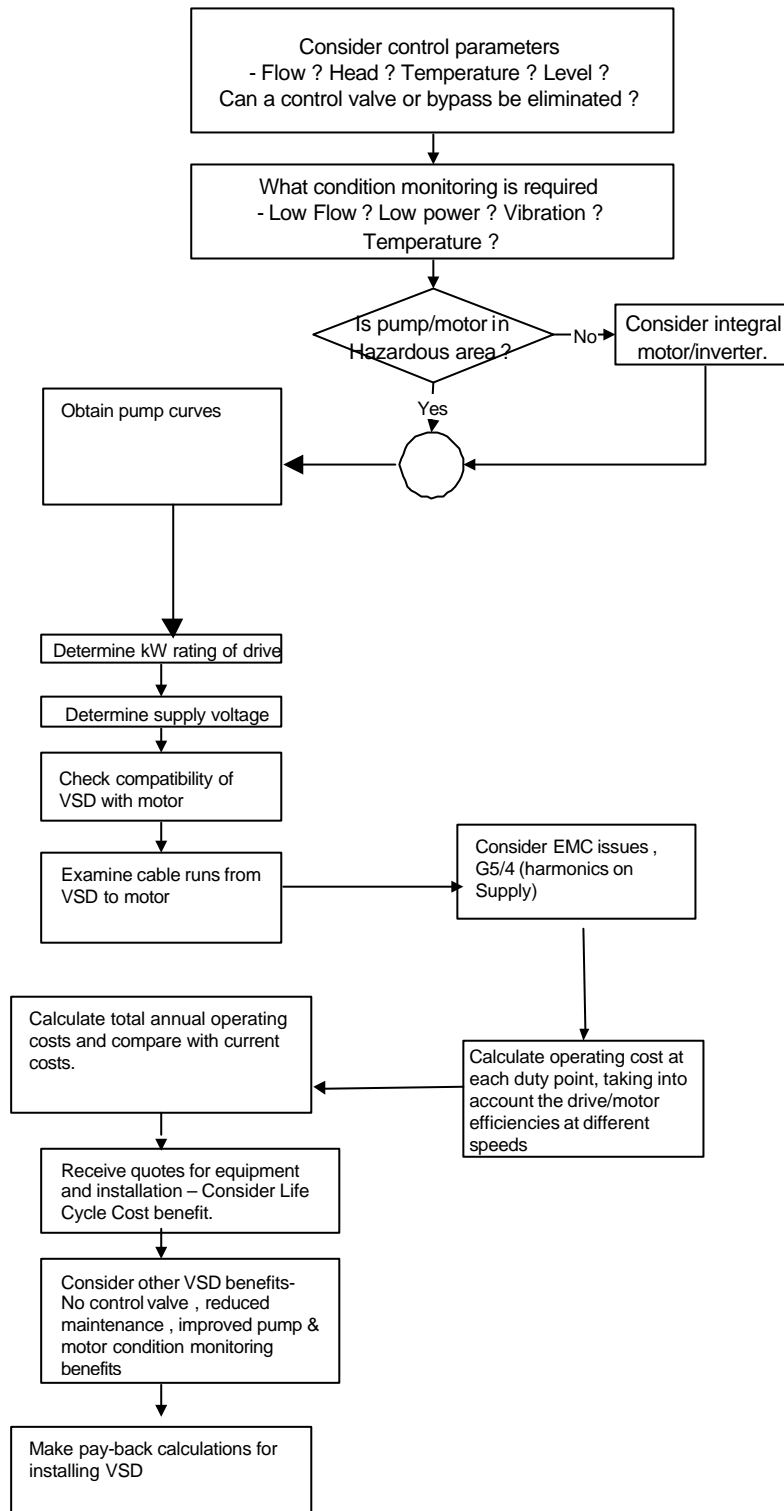
### 8.3 SIZING AND SELECTION OF A VSD ON EXISTING EQUIPMENT

To assist with decision on whether or not to retrofit on existing equipment, please refer to flow chart (B) below.

**8.3.1 Flowchart to assess the suitability of retrofitting a VSD to an existing pump system**



**Continuation of Flowchart (B) to assess the suitability of retrofitting a VSD to an existing pump system**



## **SECTION 9 EFFECTS OF NOISE & VIBRATION WHEN VARYING SPEED**

Vibration occurs in all rotating machinery when in operation. Vibrations are normally caused by imbalance of the rotating components, but can also be caused by hydraulic flow through a pump or valve. It is obviously desirable to maintain sufficiently low vibration levels in order to prevent mechanical damage. Noise is also created and is undesirable from an environmental point of view.

In centrifugal pumps, energy is transmitted via the impeller vanes to the pumping medium. Because the number of vanes is finite, pressure pulsations will arise. Flow around the vanes and flow separation can occur if the pump is operating away from its design duty, leading to an unsteady state of flow. The pressure pulsations and unsteady flow may excite vibrations in the pump casing and associated piping, with the result that noise may be transmitted to the surrounding environment.

The trend towards higher rotational speeds has an unfavourable effect on the noise developed by centrifugal pumps, since the overall dimensions of the machine are reduced and the energy conversion consequently takes place in a much smaller volume (increased power density).

It follows, therefore that a reduction in operating speed leads to lower noise and vibration levels. Since the operation of a motor driven pump in combination with a variable speed drive system, often results in lower average speeds (the unit will only operate at a speed to achieve the desired duty condition), a general reduction in noise levels is to be expected. This is, of course, a benefit where noise level is a consideration. Typical noise level restrictions in current specifications call for sound pressure levels to be below 85 dBA.

The adoption of a variable speed drive system can also lead to a lowering of hydraulic noise levels in the piping system. Additionally, the elimination of items such as control valves, will also lead to a reduction in hydraulic noise levels.

As most pumps are driven by electric motors, consideration should also be given to the noise emitted by the driver. Combined noise levels for pump/motor combinations are typically 2dBA more than the item with the higher noise level.

It should also be noted that an increase of between 3 and 6 dBA in motor noise is generally apparent when motors are used in combination with inverter drives. The overall reduction in speed level, however, normally leads to an overall decrease in noise levels.

A change in the rotating speed will vary the frequencies of the vibration exciting forces, this can create a structural resonance not found at the fixed running speed this can occur equally with reduction or increase in the speed, and is mostly associated with pumps mounted on light structures, or vertical machines which are inevitably less rigidly mounted. The manufacturer should always be consulted in these areas.

### **9.1 TYPICAL NOISE LEVELS**

#### **PUMPS**

The table below indicates the noise levels that can be expected from typical single stage centrifugal pumps operating at different speeds:



Power requirement at maximum speed (kW)	Speed (rpm)	Sound pressure level (dBA @ 1m)
3.0	2900	72
	1450	63
	980	58
37	2900	80
	1450	72
	980	67

**Table 9.1**

**MOTORS**

Small motors are relatively quiet in operation; as motor speed and size increases the noise level also increases.

It should be noted that energy efficient motors (EFF1) are quieter in operation than older designs which generally fall into the EFF3 rating – a comparison between these is shown below which is based on two sizes of typical squirrel cage motors with IP 55 enclosure.

**TYPICAL MOTOR NOISE LEVELS**

Motor Power (kW)	Motor Speed (rpm)	EFF3	EFF1
		No load Sound pressure level (dB(A) @ 1m)	
3.0	2900	78	64
	1450	58	53
	980	54	51
37	2900	85	72
	1450	76	63
	980	72	61

**Table 9.2**

## **APPENDIX - ADDITIONAL INFORMATION**

### **A1 MAGNETIC DRIVE PUMPS**

#### **A1.1 EFFECTS OF PUMPSPEED**

The affinity laws described in section 3.1.1 apply to pumps where the impeller is driven directly, with the shaft sealed by gland packing or mechanical seal where it enters the pump casing. Pumps which require a higher leak free integrity are hermetically sealed with a stationary device called a Containment Shell or Sheath. With this design, the pump shaft is driven by Magnetism across the stationary containment shell, either by the use of permanent magnets (Synchronous Drive or Canned Magnet Drive Pumps) or by the use of an induced magnetic field (Canned Motor Pumps). These react differently to speed change.

With magnetically driven pumps, there are usually magnetic power losses to be taken into account, as well as hydraulic power. These are proportional to speed squared. (The exception to this rule is that there are generally no magnetic losses incurred for non-metallic magnetically driven pumps).

Hydraulic Power follows the Affinity Laws, i.e.  $P_2 = P_1 \times (RPM_2/RPM_1)^3$   
 Magnetic Power(M) follows the physical law:-  $M_2 = M_1 \times (RPM_2/RPM_1)^2$

Then, Total Power = Hydraulic Power (P) + Magnetic Losses (M)

The magnetic power is typically 10% of the total power so the deviation caused by this departure from the Affinity Laws is only small.

NOTE:-

1. For any significant increase in speed, always verify with the pump manufacturer that the magnetic coupling is adequately rated for this new power & speed.
2. Non-metallic magnetic drive pumps usually have no magnetic losses and so power can be calculated by the affinity laws, similar to mechanically sealed pumps. However, the pump manufacturer must still always be consulted to check that the magnetic coupling is adequately rated for the new power & speed.

#### **A 1 2 EFFECTS OF IMPELLER DIAMETER CHANGE**

The equations in section 3.1.2 define the variation of pump hydraulic performance with impeller diameter for conventionally driven pumps. For Magnetically Driven pumps, the effect of diameter change on power, only applies to the Hydraulic Power. If the speed is unchanged the Magnetic losses will not change.

Using the usual hydraulic power/diameter relationship,  $P_2 = P_1 \times (D_2/D_1)^3$

Then total Power at the new diameter = Hydraulic Power ( $P_2$ ) + Constant Magnetic Losses

### **A2 MOTOR CONSIDERATIONS**

#### **PROTECTION**

As the inverter control monitors the motor current, it will also provide motor protection, based on the load and speed, and compensate for reduced loadability at reduced speed.

Some more sophisticated drives offer both over and under load protection, which allows monitoring of imminent seizure, coupling failure, low flow and dry run.

## **SPEED**

A motor is not limited to its synchronous speed; it can be both reduced and increased. Figure 8.1 shows that beyond base speed the torque available will drop as the inverter cannot increase its output voltage, and the motor becomes progressively under fluxed. This is known as field weakening

There is a specific maximum output speed for any motor, and the advice of the manufacturer should be sought before running a motor beyond its base speed.

## **INSULATION DESIGN**

Modern motor insulation systems are suitable for inverter use; however, the manufacturer should be asked to confirm. Further information is available from the Gambica/REMA Technical Report No 1<sup>3</sup>

## **MOTOR BEARINGS**

Motor bearings may run hotter with an inverter fed motor, as some additional losses will occur in the rotor, and are dissipated down the shaft, potentially requiring high temperature lubricants, or more frequent lubrication. In larger motors (greater than 100kW) electric discharge through bearings may require insulated bearings to be fitted. Further information is available from the Gambica/REMA Technical Report No 2<sup>4</sup>

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<sup>3</sup> See Appendix A6

<sup>4</sup> See Appendix A6

## **A3 LEGISLATIVE REQUIREMENTS**

Within Europe the inter-relation of a system with its environment is laid down by a number of EU Directives, and by local regulation, these are direct obligations on users, and their agents to be applied when implementing a system.

A number of EU Directives are particularly important when considering variable speed pumping applications. EU Directives are implemented into UK law by parliamentary Statutory Instruments.

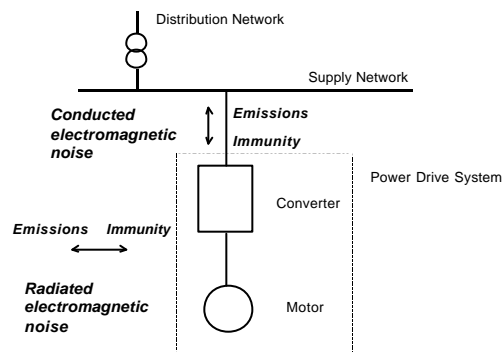
### **A3.1 THE MACHINERY DIRECTIVE<sup>5</sup>**

The Directive is implemented by listing a number of harmonised standards, which must be applied.

From the point of view of an electrically driven pump the most important of these standards is likely to be EN 60204-1

### **A3.2 THE EMC DIRECTIVE<sup>6</sup>**

EMC is the ability of a device or system to function without error in its intended electromagnetic environment without disturbing other devices in that environment.



**Figure A3.1 –**

### **EMC aspects of a Power Drive System**

The EMC Directive details the levels of electrical disturbances, both radiated and conducted that the drive system is permitted to emit and also the levels to which it is immune. In general the immunity limits also appear within the scope of the Machinery Directive, and the Low Voltage Directive.

<sup>5</sup> Directive 98/37/EC published by the EU Commission, Rue de la Loi 200, B-1049, Brussels.

<sup>6</sup> Directive 89/336/EEC, modified by 91/263/EEC and 93/68/EEC published by the EU Commission, Rue de la Loi 200, B-1049, Brussels.

### **A3.2.1 RADIATED EMISSIONS**

The drive is a source of high frequencies due to the controlling microprocessors, and the rapidly rising wave fronts of the output voltage pulses.

Unsuitable cabling and incorrect grounding can allow these high frequencies to radiate and disturb the environment.

### **A3.2.2 CONDUCTED EMISSIONS**

Lower frequency disturbances can be passed back through the supply network to cause problems for other users of that network.

To meet the requirements of the Directive a “Power Drive System” must comply with the harmonised Product Specific Standard EN61800-3.

To avoid problems, it is essential that the manufacturers installation recommendations be strictly followed.

### **A3.2.3 HARMONICS**

Another aspect of EMC is low frequency emissions, or harmonics. The level of harmonics reflected back to the supply network is usually regulated by the electricity supply utility.

Harmonics are voltages and currents in the electrical system at frequencies that are multiples of the fundamental frequency (50 Hz in UK power systems).

Harmonics are associated with any load that uses a rectifier based power supply such as radio and TV, computers, lighting ballasts, other domestic equipment such as washing machines and microwave ovens etc.

Harmonics also come from generators, and transmission equipment. They can affect the equipment performance and are both caused by and can interfere with the function of VSDs.<sup>78</sup>

### **A3.3 THE LOW VOLTAGE DIRECTIVE<sup>9</sup>**

This Directive concerns all electrical equipment with nominal voltages from 50 V to 1000 V AC (and 75 V to 1500 V DC).

The aim is to provide protection against electrical, mechanical fire and radiation hazards.

The Low Voltage Directive does not apply to products for use in potentially explosive atmospheres.

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<sup>7</sup> Harmonics in the UK are governed by the Electricity Association Engineering Recommendation G5/4, published by The Electricity Association, Millbank, London.

<sup>8</sup> Information on compliance is contained in Gambica Technical Guide No 1.

<sup>9</sup> Directive 72/23/EEC, modified by 93/68/EEC published by the EU Commission, Rue de la Loi 200, B-1049, Brussels.

### **A3.4 THE ATEX DIRECTIVE<sup>10</sup>**

This Directive provides a framework for determining the essential health and safety requirements for use of any product in a potentially explosive atmosphere.

The Directive applies to the entire equipment INCLUDING the pump. It also includes operation within dust hazards, as well as flammable gases.

Under the terms of the directive it is unlikely that any electric motor manufactured under previous standards, or meeting earlier versions of this directive will be permitted to be retrofitted with a variable speed drive.

When considering an installation in a hazardous area the advice of the motor and converter manufacturer must be sought to enable a properly matched and certified system to be achieved.

### **A3.5 THE CE MARKING DIRECTIVE<sup>11</sup>**

Under the terms of the EU Directives it is a statutory requirement that all electrical and electronic systems installed in Europe meet the requirements of all appropriate directives, and carry the appropriate “CE” compliance indication.

For all equipment covered by EU Directives the principal document showing conformity is the Manufacturers’ Declaration of Conformity. The pumping system should not be commissioned without the appropriate declarations to the EMC, Low Voltage and Machinery Directives, plus the ATEX Directive if applicable, being available.

## **A4 ABBREVIATIONS**

### **PUMP**

NPIP	Net positive inlet pressure
NPSP	Net positive suction pressure
NPSH	Net positive suction head
NPSHA	Net positive suction head available
NPSHR	Net positive suction head required
H	Head
Q	Flow
Rpm	Revolutions per minute
PD	Positive displacement pump
BEP	Best efficiency point

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<sup>10</sup> Directive 94/9/EC (to be fully implemented from 1<sup>st</sup>. July 2003) published by the EU Commission, Rue de la Loi 200, B-1049, Brussels.

<sup>11</sup> Directive 93/68/EEC published by the EU Commission, Rue de la Loi 200, B-1049, Brussels.

## **MOTOR AND DRIVE**

HEM	High efficiency motor
PWM	Pulse width modulated
AC	Alternating current (voltage)
DC	Direct current (voltage)
EMC	Electromagnetic Compatibility
PLC	Programmable logic controller
VSD	Variable speed drive
VFD	Variable frequency drive
VVVF	Variable voltage variable frequency (drive)
TEFV	Totally enclosed fan ventilated

## **A5 REFERENCES AND FURTHER READING**

GAMBICA/REMA Technical Report No 1. Motor insulation voltage stresses under PWM inverter operation  
GAMBICA/REMA Technical Report No 2 Motor shaft voltages and bearing currents  
GAMBICA Technical Guide Managing Harmonics, A guide to EA Engineering Recommendation G5/4  
Both may be freely downloaded from the appropriate organisation website (See below)  
NEMA Standards Publication - Application Guide For AC Adjustable Speed Drive Systems May be freely downloaded from the NEMA website (See below)  
Pump Life Cycle Costs - by Europump & Hydraulic Institute, available from BPMA website ( See below)  
Energy Efficiency report, 1987, Greenwood PB  
Energy Efficiency of Electric Motors by Design, 1994, Walters DG & Williams IJ  
Good Practice Guide 2, Energy Savings with Electric Motors and Drives, 1998, Active Energy  
Good Practice Guide 49, Energy Savings in Pumping Systems, 1998, Active Energy  
General Information Leaflet 56, Implementing a motor management policy, Active Energy  
Gambica /REMA guide 'Making The Most of The Climate Change Levy Package'  
European motor manufacturers catalogues.  
European Pumps and Pumping by McGraw Hill  
Pumping Handbook by McGraw Hill  
AEMT Good Practice Guide, The repair of Induction Motors.  
Good Practice Guide 249, Energy Savings in Industrial Water Pumping Systems - Action Energy  
Best practice Case Studies, Hundreds of actual energy saving examples are detailed in these case studies - Action Energy.  
Action Energy makes available a huge volume of detailed guidance including Best Practice guides and Case Studies. These are available on the web ([www.actionenergy.org.uk](http://www.actionenergy.org.uk)) or as hard copies from the helpline (0800 585794)

Websites:-  
[www.gambica.org.uk](http://www.gambica.org.uk)  
[www.rema.uk.com](http://www.rema.uk.com)  
[www.nema.org](http://www.nema.org)  
[www.bpma.org.uk](http://www.bpma.org.uk)  
[www.actionenergy.org.uk](http://www.actionenergy.org.uk)

VARIABLE SPEED DRIVEN PUMPS-BEST PRACTICE GUIDE.

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