Flow of liquids

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3.1 Fundamental equations

3.1.1 Explanation of concepts

The processes of flow occurring in nature are usually very complex and difficult to deal with. In many technical applications, however, perfectly acceptable results can be obtained from calculations based upon simplified considerations. Some of the concepts and conditions germane to this concept are discussed in this chapter.

A flow process is steady if all flow parameters - pressure, velocity, etc. - at any specific point in the flow field are independent of time. According to this definition practically all flow processes are unsteady. In practice, many can, with sufficient accuracy, be treated as steady using suitable mean time values. The fundamental equations presented in later sections are valid for steady flow.

Extra care should be taken with certain pumping installations where steady state conditions may not exist. Low head, high flow centrifugal pumps with wide impeller tip widths produce velocity variations across the flow stream. Some positive displacement pumps, with a small number of pumping elements, produce cyclic flow and pressure variations. The amplitude of these variations is dependant upon the piping system as well as the pump construction.

Flow is generally three-dimensional, i.e. the flow parameters vary with all three co-ordinates defining some point in space. In many practical cases of flow the number of dimensions studied may be reduced whilst still maintaining accuracy. A common example is pipe flow, where the parameters are assumed to vary in only one direction, i.e. the longitudinal dimension of the pipe.

One-dimensional flow assumes that flow parameters can be described by mean values across the flow through-section, see Figure 3.1. In principle, different mean values should be applied when studying continuity, momentum and energy. In the case of pipe flow, the mean velocity in the pipe is defined as the volume flow divided by the cross-sectional area of the pipe (v = Q/A). This mean velocity can as a rule be used with adequate accuracy for most purposes.

An important property of a flowing medium is its density and the changes in density which occur during flow. A gas is compressed, i.e. the density increases, when the pressure somewhere in the flow increases. This type of flow is called compressible flow. In a liquid system density variations are very small under great changes of pressure. Liquid flow can thus, with sufficient accuracy, be treated as incompressible. This is also the case with gases at low flow velocities when the pressure changes are small. Liquid compressibility may be important for pump selection. Hot liquids and liquefied gases may require special consideration depending upon the differential pressure. Power is required to compress liquids. Ignoring liquid compressibility can result in low flow and/or undersized pump drivers.

A streamline is a curve to which the velocity vector is tangential at any point. In the case of steady flow, the streamlines remain unchanged with time and describe the path of a liquid particle passing through the flow field. The streamlines through all points on a closed curve in the flow field constitute a flow tube. See Figure 3.2. No mass will pass through the circumferential boundary surface of a flow tube. The flow tube is thus reminiscent of an ordinary pipe. In the case of a pipe, however, there are strong frictional effects associated with the wall of the pipe which do not necessarily occur in a flow tube.

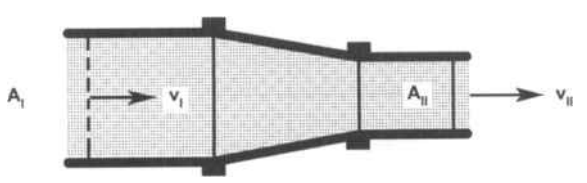

Figure 3.1 Example of one-dimensional flow

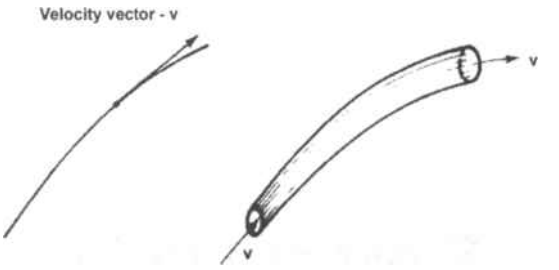

Figure 3.2 Streamline and flow tube

3.1.2 The continuity equation

The continuity equation is a statement for the condition that mass is not created or destroyed during a flow process.

Assuming that the flow is steady, the mass flow $\dot{m}$ must be of equal magnitude everywhere along the pipe or the flow tube. In the case of one-dimensional flow in Figure 3.1.

$$\dot{m} = r_I \cdot v_I \cdot A_I = r_{II} \cdot v_{II} \cdot A_{II} \qquad \text{Equ 3.1}$$

or for an incompressible liquid flow,

$$Q = v_I \cdot A_I = v_{II} \cdot A_{II} \qquad \text{Equ 3.2}$$

where:

Q = volume flow (m^3/s)

v = Q/A = flow velocity (m/s)

A = cross-sectional area (m^2)

When the cross-sectional area in a pipe reduces, then, according to the continuity equation, the flow velocity increases. If the area is halved, the velocity doubles and so on.

Since there is no increase in mass at the point of branching, the mass flow entering will equal the total mass discharging per unit time. Using the symbols of Figure 3.3,

$$Q_I = Q_{II} + Q_{III}$$

or

$$v_I \cdot A_I = v_{II} \cdot A_{II} + v_{III} \cdot A_{III} \qquad \text{Equ 3.3}$$

3.1.3 Bernoulli's equation

Bernoulli's equation is an equation of motion for incompressible flow based on Newton's second law of motion adapted for fluids and Euler's equation of motion. The equation expresses the relationship between:

- static pressure
- potential energy
- kinetic energy

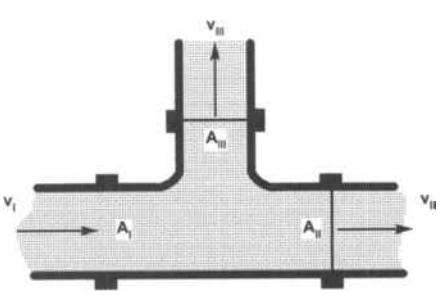

Figure 3.3 Branching

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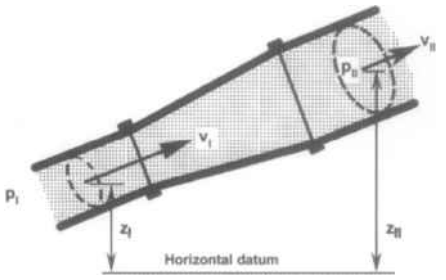

Figure 3.4 Symbols used in Bernoulli's equation

- friction losses

Using Figure 3.4, Bernoulli's equation, for steady, one-dimensional, constant temperature and incompressible flow between stations I and II, becomes:

$$\frac{p_I}{\rho} + \frac{v_I^2}{2} + gz_I = \frac{p_{II}}{\rho} + \frac{v_{II}^2}{2} + gz_{II} + \frac{\Delta p_f}{\rho} \qquad \text{Equ 3.4}$$

where:

p	=	static pressure (gauge) (Pa)
ρ	=	density (kg/m^3)
v	=	velocity (m/s)
g	=	gravitational acceleration (m/s^2)
z	=	height above datum (m)
Δp_f	=	frictional pressure loss (Pa)

Equation 3.4 expresses the various energy terms per unit mass of fluid, J/kg.

In rotodynamic pump technology it is practical to express Bernoulli's equation in terms of "head" i.e. metres of liquid column. If all terms in Bernoulli's equation are divided through by g:

$$\frac{p_I}{w} + \frac{v_I^2}{2g} + z_I = \frac{p_{II}}{w} + \frac{v_{II}^2}{2g} + z_{II} + \frac{\Delta p_f}{w} \qquad \text{Equ 3.5}$$

where:

w	=	specific weight (N/m^3)
w	=	ρg

The various terms are then called:

$\frac{p}{rg}$ or $\frac{p}{w}$ = static head (m)

$\frac{v^2}{2g}$ = velocity head (m)

z = potential head (m)

$\frac{\Delta p_f}{w}$ = h_f = head loss (m))

Since all the terms in equation 3.5 refer to height they are easy to illustrate graphically. See Figure 3.5.

Total head is sometimes designated by H. The relationship between pressure and head is p/w = h. The three head terms can be converted to equivalent pressure terms. Static pressure plus velocity, or dynamic pressure is called total pressure or stagnation pressure.

$$\text{total pressure} = p + \frac{rv^2}{2} = wh + \frac{wv^2}{2} \qquad \text{Equ 3.6}$$

Total and static pressure are measured in different ways and illustrated in Figure 3.6. A pressure tap at right angles to the direction of flow will sense the static pressure. Ahead of a Pitot tube the velocity is reduced to zero and the static pressure rises to stagnation pressure.

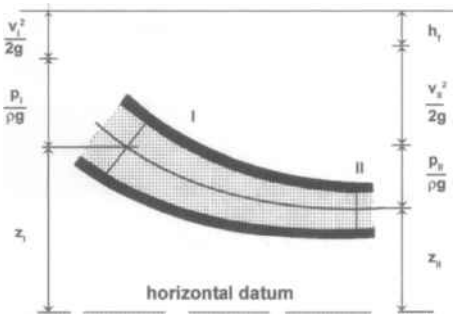

Figure 3.5 Graphic illustration Bernoulli's equation

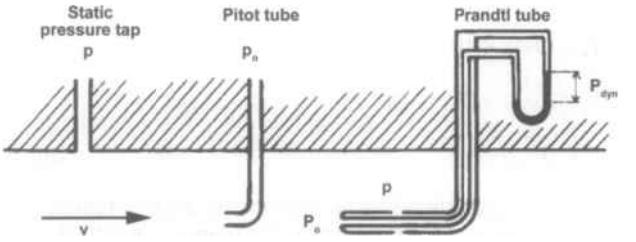

Figure 3.6 Measuring of static, total and dynamic pressure

As a rule, it is easy to measure the static pressure, whereas the total pressure can easily be affected by measurement errors. A Prandtl tube or Pitot-static tube measures both total and static pressure. The differential pressure, the dynamic pressure, $p_{dyn} = wv^2/2g$ is obtained directly.

3.1.4 The momentum equation

The product of mass times flow velocity for a liquid particle is called its momentum. The momentum equation for steady flow reads:

$$F = \frac{d}{dt} \cdot \dot{m}$$

The resultant F of all external forces on a control volume is equal to the changes in momentum for the outflowing and inflowing mass per unit time, or,

$$F = \dot{m}\left(\bar{v}_2 - \bar{v}_I\right) \qquad \text{Equ 3.7}$$

where:

$\dot{m}$	=	mass flow dm/dt $\left(\frac{kg/s}{(m/s)}\right)$
$\bar{v}$	=	velocity vector

The momentum equation is illustrated in the following example. The problem is to determine the force required to hold a pipe bend in position, as illustrated in Figure 3.7.

The control volume is defined by the interior of the pipe bend.

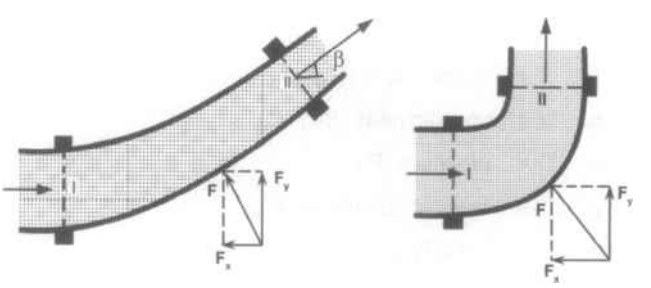

Figure 3.7 Pipe bends, arbitrary (left) and 90° bend

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Then, for the arbitrary bend in the x axis:

$$-F_x + p_I \cdot A_I - p_{amb} \cdot A_I - p_{II} \cdot A_{II} \cdot \cos\beta + p_{amb} \cdot A_{II} \cdot \cos\beta = \dot{m}\left(v_{II} \cdot \cos\beta - v_I\right)$$

or:

$$F_x = \dot{m} \cdot v_I + \left(p_I - p_{amb}\right) \cdot A_I - \left[\dot{m} \cdot v_{II} + \left(p_{II} - p_{amb}\right) \cdot A_{II}\right] \cdot \cos\beta$$

Similarly, in the y axis:

$$F_y = \left[\dot{m} \cdot v_{II} + \left(p_{II} - p_{amb}\right) \cdot A_{II}\right] \cdot \sin\beta$$

For a 90° bend $\beta = 90°$, therefore:

$$F_x = \dot{m} \cdot v_I + \left(p_I - p_{amb}\right) \cdot A_I$$

$$F_y = \dot{m} \cdot v_{II} + \left(p_{II} - p_{amb}\right) \cdot A_{II}$$

$$F = \sqrt{F_x{}^2 + F_y{}^2} \qquad \text{Equ 3.8}$$

NOTE: It is a gauge pressure and not the absolute pressure in the pipe which determines the magnitude of the force.

Also that the forces can be determined without detailed knowledge of the internal flow process through the bend. The momentum equation applies regardless of whether the process has losses or not.

It is often the case that the terms m and v are small compared with the others. For a 90° bend with A_I being equal to A_{II}.

$$F = \sqrt{2} \cdot \left(p - p_{amb}\right) \cdot A \qquad \text{Equ 3.9}$$

where:

p	=	pressure in pipe (absolute) (Pa)
p_{amb}	=	ambient pressure (absolute) (Pa)
A	=	pipe cross-sectional area (m^2)
F	=	fixing force (N)

The fixing force has an angle of 135° to the incoming direction of flow.

3.1.5 The energy equation

The energy equation is an extension of the energy principle which states that energy cannot be created or destroyed but can only be converted to other forms. The energy equation for steady state one-dimensional pump flow is:

$$W_{in} = \left(h_{II} - h_I\right) + \frac{\left(c_{II}^2 - c_I^2\right)}{2} + g\left(z_{II} - z_I\right) + Q_{out} \qquad \text{Equ 3.10}$$

The term h, specific enthalpy, is made up from two components; u, ($u = Tc_p$) internal energy and pv, which represents the mechanical energy due to pressure, often called flow work. The energy equation can be rewritten:

$$W_{in} = \left(T_{II}c_{p_{II}} - T_I c_{p_I}\right) + \left(p_{II}v_{II} - p_I v_I\right) + \left(\frac{c_{II}^2 - c_I^2}{2}\right) + g\left(z_{II} - z_I\right) + Q_{out} \qquad \text{Equ 3.11}$$

where:

T	=	temperature (°C)
c_p	=	specific heat (J/kg/°C)
p	=	pressure (Pa)
v	=	specific volume (m^3/kg)
c	=	velocity (m/s)
z	=	height above datum (m)
g	=	gravitational acceleration (m/s^2)

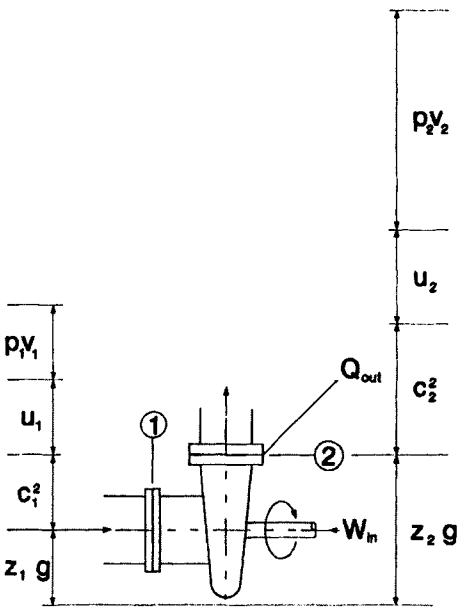

Figure 3.8 Illustration of the energy equation

W_{in}	=	work input to the shaft (J/kg)
Q_{out}	=	heat rejected (J/kg)

NOTE: In these equations the velocity is represented by c not v. Engineers frequently use the same symbol for different quantities. The context is important and should be explanatory. All the terms in the energy equation are calculated per kg of medium flowing. The terms $c^2/2$ and gz respectively represent the kinetic and potential energy of the medium per kg.

The effect of a pump on the energy equation is shown diagrammatically in Figure 3.8. The input of mechanical work is through the rotating shaft, all the variables change and some heat is rejected from the pump body. Notice particularly that a temperature rise has occurred. The change from suction conditions to discharge conditions usually takes place very quickly within the pump so the ideal process is isentropic, constant entropy, not constant temperature. The isentropic temperature rise is small for water, of the order of 0.4 °C per 100 bar pressure rise at 50°C rising to 2.6°C per 100 bar rise at 250°C.

Pump inefficiencies will generally appear as increased temperature rise and heat rejection. Noise and vibration will take their energy as portions of heat rejection. In equations 3.10 and 3.11, g is the only constant, for the location, not universally. Even c_p is a variable, from less than 4.0 to over 5.0 for water.

Rotodynamic pump designers take the centre-line of the first stage impeller eye as the datum for all calculations and parameters such as NPSHr. Most positive displacement pump designers take the centre-line of the inlet connection as the datum. A system designer will not know precisely where the impeller eye or inlet connection will be. He should base all the system calculations on the foundation level where the pump will be located or the liquid surface level for submerged pumps.

The quality of any calculated result is dependant upon the accuracy of the input data. Pumps always operate in the pressurised liquid region so that saturated liquid data will not be accurate. Steam tables, such as the UK Steam Tables, based on the 1967 IFC Formulation, contain tabulated data for pressurised water up to 1000 bara. However this data is aimed at the power generating industry and is tailored to their needs. Inconsistencies exist in the data for low temperatures at pressures close to atmospheric. This data should be examined carefully before use and "conditioned" if thought appropriate. Unconditional use of raw data can result in errors of up to 6% when compared to results obtained from calculations in Sections 3.2 and 3.4.

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Computerised steam tables alleviate some of the inconsistencies.

In some cases it will be possible to ignore the effects of changes in velocity, specific volume and height, between inlet and outlet, without loss of accuracy. Pump differential head or pressure is the important criterion. No assumptions should be made on pumps which operate at low differentials, below 50m.

3.2 Pipe flow losses

3.2.1 Reynolds Number

Flow losses occur because of the effect of internal friction. Shear stresses arise as soon as velocity gradients exist.

$$\tau = \mu \cdot \frac{dv}{dy} \qquad \text{Equ 3.12}$$

where:

τ = shear stress (N/m^2)

μ = viscosity (Ns/m^2)

v = flow velocity (m/s)

y = thickness of flow stream(m)

The work done by the shear force generates heat and adds to the internal energy of the liquid. The increase in internal energy means that the static pressure becomes a little less, hence the name pressure loss or pressure drop. The level to which the liquid temperature can rise due to friction heating is dependant upon the heat transfer mechanism to the surroundings.

The flow losses are great wherever the shear stress is great, i.e. where the velocity gradients are great. High velocity gradients occur in the boundary layer, at flow around sharp corners, in strong vortex flows and so on. Viscous flow in pipes is characterised by Reynolds Number.

$$Re = \frac{v \cdot d}{\nu} \qquad \text{Equ 3.13}$$

where:

Re = Reynolds Number (dimensionless)

v = flow velocity (m/s)

d = pipe diameter (m)

ν = kinematic viscosity (m^2/s)

The following relationship exists between kinematic viscosity, ν, and dynamic viscosity, μ.

$$\nu = \frac{\mu}{\rho} \qquad \text{Equ 3.14}$$

where:

ρ = liquid density (kg/m^3)

In the case of low Reynolds Numbers, Re <2300, the flow is laminar. The liquid flows in layers with different velocities which do not mix with each other. At Re > 2300 the flow is turbulent, that is to say the liquid particles do not flow in a straight line in the direction of motion of the main flow.

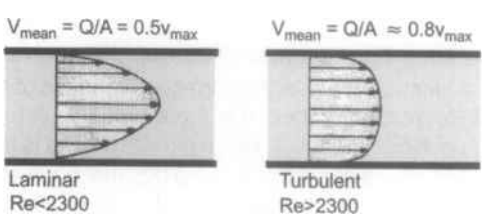

Figure 3.9 Laminar and turbulent flow

Since the velocity gradients differ, the shear stress and flow losses will also differ between the two cases. The critical value of Re Number has been stated as Re cr ≅ 2300. It should be noted however, that the flow may continue to be laminar at considerably higher values of Re if the flow is very well protected from disturbances and entrance conditions to the section are the optimum. Low values of Re Number occur at low flow velocities, in small pipe diameters or at high viscosity. See Figure 3.9.

Liquid	Kinematic Viscosity ν m^2/s	Transition to turbulent flow (for the pipe diameters given) Vcr m/s			
		d = 0.001m	d = 0.01 m	d = 0.1 m	d = 1 m
Water, 20°C	$1 \cdot 10^{-6}$	2.3	0.23	0.023	0.0023
SAE 5W 60°C	$1 \cdot 10^{-5}$	23	2.3	0.23	0.023
Burner oil Class E (BS 2869: 1970) 30°C	$1 \cdot 10^{-4}$	230	23	2.3	0.23
SAE 50 18°C	$1 \cdot 10^{-3}$	2300	230	23	2.3

Table 3.1 Transition to turbulent flow in pipes

It can be seen from the examples, see Table 3.1, that laminar flow is less usual in practice for liquids with viscosity similar to that of water, but is quite common for oils.

3.2.2 Head losses in straight pipes

The following formula applies to head losses for flow in a straight pipe.

$$h_f = \lambda \cdot \frac{l}{d} \cdot \frac{v^2}{2g} (m) \qquad \text{Equ 3.15}$$

where:

λ = loss coefficient for straight pipes (dimensionless)

l = length of pipe (m)

d = pipe diameter (m)

v = Q/A = flow velocity (m/s)

The loss coefficient is dependent upon Reynolds Number and upon the internal roughness of the pipe. This is shown in Figure 3.10.

Pipe and pipe material	Condition of pipe	Factor k Calculation values mm
Steel pipe	Seamless, new	0.03 - 0.05
Steel pipe	Welded, new	0.03 - 0.10
Steel pipe	In use, rusted	0.15 - 0.5
Steel pipe	Seamless, galv.	0.1 - 0.16
Steel pipe	Welded, galv.	0.1 - 0.2
Steel pipe	Heavy asphalt coating	0.5
	Mature water mains	1.2
	Stainless, acid resist.	0.045
Cast iron pipe	Bitumen coated	0.1 - 0.3
	Main conduit	0.1
	Other conduits	0.2 - 0.3
	In use, rusted	0.4 - 2.0
Copper pipe	Diameter <200 mm	0.01
Brass, glass	Technically smooth	0.002
Concrete pipe	Water main	0.1 - 0.2
	Waste, drains, new	0.2 - 0.5
	Waste, drains, old	1.0
Asbestos cement pipe	Newly laid	0.05
Earthenware pipe	Newly laid drains	0.2
	Older drains	1.0
	Mature foul sewer	3.0

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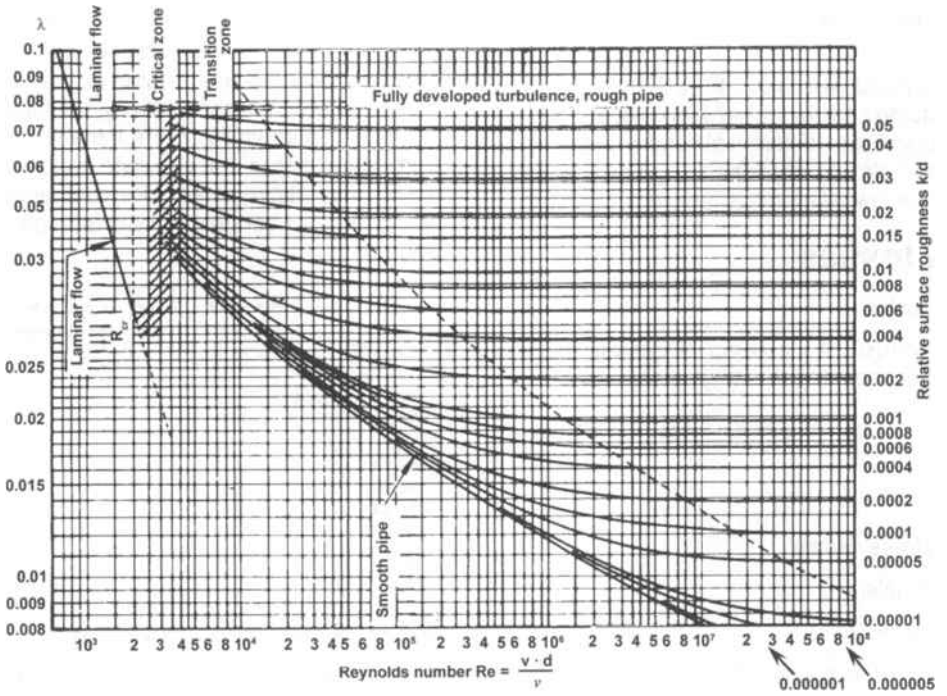

Figure 3.10 Loss coefficient for straight pipes

Pipe and pipe material	Condition of pipe	Factor k
		Calculation values mm
PE, PVC	Dia. <200> mm	0.01
PE, PVC	Dia. 200 mm	0.05
GRP	All dimensions	0.1
Wood	New	0.2 - 1.0

Table 3.2 Examples of approximate roughness in pipes

The roughness of the pipe can be assessed from the values in Table 3.2.

The loss coefficient can also be calculated from the following formulae:

Laminar flow Re < 2300

$$\lambda = \frac{64}{Re} \qquad \text{Equ 3.16}$$

Turbulent flow Re > 4000

$$\frac{1}{\sqrt{\lambda}} = -2 \cdot \log_{10}\left(\frac{2.51}{Re\sqrt{\lambda}} + \frac{k/d}{3.71}\right) \qquad \text{Equ 3.17}$$

Equation 3.17 is sometimes called the Colebrook-White equation; Colebrook in France and Prandtl-Colebrook in Germany. In the region Re = 2300 to Re = 4000 laminar and turbulent flows may alternate in various sections of the pipe. The resultant loss coefficient will then assume values between those obtained in formulae 3.16 and 3.17.

The expression given in equation 3.17 is somewhat inconvenient for manual calculation. The following formula, proposed by Miller, can be used for turbulent flow:

$$\lambda = \frac{0.25}{\left[\log_{10}\left(\frac{k}{3700d} + \frac{5.74}{Re^{0.9}}\right)\right]^2} \qquad \text{Equ 3.18}$$

Other formulae used for the calculation of straight pipe losses in the turbulent region are the Hazen-Williams and Manning formulae. These, however, do not provide any additional information to the equations presented above and have, therefore, been omitted.

3.2.3 Head losses in fittings

Head losses in bends and valves, etc. can be calculated with the help of the formula:

$$h_f = \varsigma \cdot \frac{v^2}{2g} \text{ (m)} \qquad \text{Equ 3.19}$$

where:

ζ = loss coefficient of fitting (dimensionless)

v = flow velocity (m/s)

The magnitude of the head losses are in principle influenced, as in the case of straight pipe flow, by surface roughness and by Reynolds Number.

Examples of approximate values of loss coefficients for losses in fittings are illustrated in Figure 3.11. All values are those for conditions of normal roughness and at high Reynolds Number, i.e. at fully developed turbulent flow. In cases where the velocity changes it must always be the highest velocity which is used for calculating h_f in accordance with equation 3.19. In the case of varying diameters, the velocity to be applied is the one resulting from the smallest diameter, and in the case of T-pieces, the velocity related to the total volume flow is applicable.

An alternative method of expressing the magnitude of head losses in fittings is the use of the concept of equivalent pipe length, l_{eq}, i.e. the equivalent length, (l_{eq}) of straight pipe giving rise to the same losses in pressure as the fitting, at the same flow velocity. By comparing equations 3.15 and 3.19:

$$l_{eq} = \frac{\zeta}{\lambda} \cdot d \qquad \text{Equ 3.20}$$

where:

ζ = loss coefficient of fitting (dimensionless)

λ = loss coefficient for straight pipes (dimensionless)

It can be seen from equation 3.20 that the equivalent pipe length is a function of λ, i.e. the surface roughness of a straight pipe and Reynolds Number. It is therefore difficult to quantify l_{eq}, as a specific property for, say, a pipe bend and is thus of an approximate character. For λ = 0.02, the equivalent pipe length of a short radius 90° bend (r = d) is l_{eq} = 20d.

For additional information and data on losses see Section 3.6.

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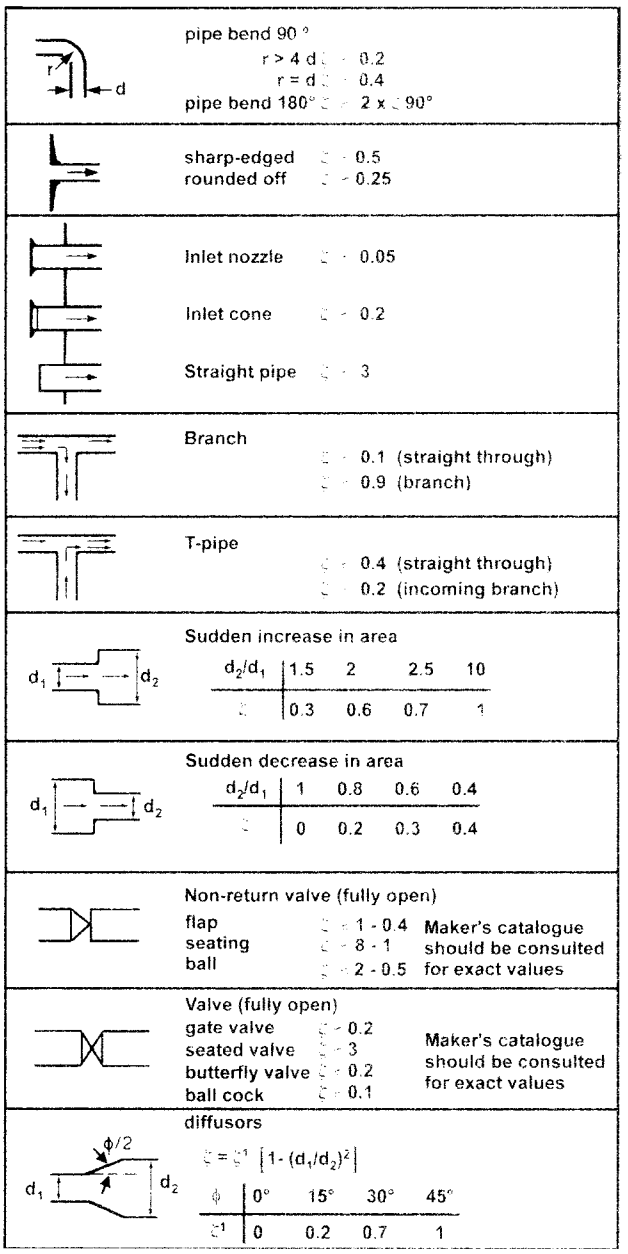


Figure 3.11 Approximate values for loss coefficients for head losses in fittings
The values for flow meters are stated in Section 3.5

3.2.4 Hydraulic diameter

For the case of flow in a partially filled pipe, in non-cylindrical ducts or in open channels, the head losses can in principle be calculated in the same way as described in earlier Sections. However, instead of using the diameter of the pipe, the hydraulic diameter, d_h, must be used in these cases.

$$d_h = \frac{4 \cdot A}{P} \qquad \text{Equ 3.21}$$

where:

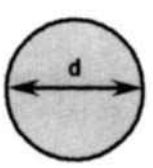
d_h = hydraulic diameter (m)

A = cross-sectional area flowing (m^2)

P = wetted perimeter (m)

For a completely full cylindrical pipe:

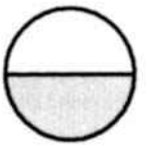
$$d_h = \frac{4 \frac{\pi \cdot d^2}{4}}{\pi \cdot d} = d$$



i.e. in this case the hydraulic and geometric diameters coincide.

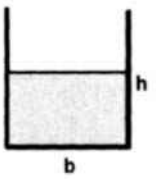
For a half-filled cylindrical pipe:

$$d_h = \frac{4 \cdot \frac{\pi \cdot d^2}{2 \cdot 4}}{\frac{\pi \cdot d}{2}} = d$$



For a half-filled rectangular section:

$$d_h = \frac{4 \cdot b \cdot \frac{h}{2}}{b + 2 \cdot \frac{h}{2}} = \frac{2 \cdot b \cdot h}{b + h}$$



With Reynolds Number

$$Re = \frac{v \cdot d_h}{\nu} \qquad \text{Equ 3.22}$$

and relative roughness k/d_h, Figure 3.10 and equations 3.16 and 3.18 apply with unchanged numerical values. The value of Re_{cr} will change as the section proportions change and will have to be evaluated by experiment or computer simulation.

Pipes flowing entirely due to gravitational head can operate completely filled. The degree of fullness depends, among other things, on the slope of the pipe and the operating conditions. For a pipe the fullness is defined by the ratio d^1/d where d^1 is the depth of liquid in the pipe and d is the diameter of the pipe. Figure 3.12 illustrates how various flow parameters change with the degree of fullness of the pipe. Both flow velocity and hydraulic diameter have their greatest value at a filling coefficient of just below 1.

For gravity flow pipes carrying solids and liquids, care must be taken with regard to sedimentation; the avoidance of which can place restrictions on the viable configuration and operation of the pipe.

Settling liquid-solid mixtures must be transported at a velocity greater than the critical velocity. If the flow velocity is reduced below the critical velocity sedimentation occurs. Sedimentation continues until the flow velocity is increased to the critical velocity. The flow area of the pipe is reduced until the mixture flow velocity is high enough to prevent further sedimentation. The hydraulic characteristic of the pipe is modified because the shape is no longer circular.

The hydraulic diameter of the flow regime can be estimated from Figure 3.13. The reduced hydraulic diameter can be used to estimate the friction loss in the partially blocked pipe. This can only be an **estimate.** The pipe wall may be smooth and well-defined. The surface of the sediment may not be smooth and the roughness may be difficult to approximate. The surface may also be moving if the particles are small. Figure 3.13 can also be used for hydraulic diameters for partially-round pipes in special applications.

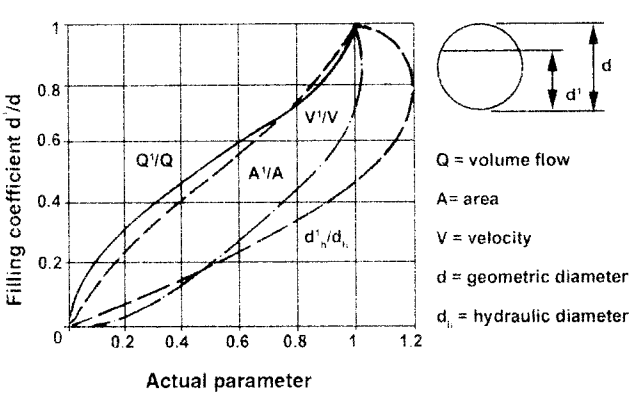


Figure 3.12 Flow in a partially-filled pipe

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3.2.5 Total head losses in the pipe system

The system total losses consist of the sum of the straight pipe head losses, the head losses in fittings and valves etc. and static losses due to changes in elevation.

$$h_f = h_f(\text{straight pipe}) + h_f(\text{fittings\& valves}) + \Delta z \quad \text{Equ 3.23}$$

With the usual symbols, head loss

$$h_f = \lambda \cdot \frac{1}{d} \cdot \frac{v^2}{2g} + \Sigma\zeta \cdot \frac{v^2}{2g} + \Delta z$$

$$= \left(\lambda \cdot \frac{1}{d} + \Sigma\zeta\right) \cdot \frac{v^2}{2g} + \Delta z$$

$$= \left(\lambda \cdot \frac{1}{d} + \Sigma\zeta\right) \cdot \frac{Q^2}{\left(\frac{\pi \cdot d^2}{4}\right)^2} \cdot \frac{1}{2g} + \Delta z \quad \text{Equ 3.24}$$

where:

h_f	=	head loss (m)
λ	=	loss coefficient for straight pipe
ζ	=	loss coefficient for fittings, valves etc.
$\Sigma\zeta$	=	sum of all loss coefficients
l	=	length of pipe (m)
d	=	diameter of pipe (m)
Q	=	volume flow (m^3/s)
v	=	flow velocity (m/s)
Δz	=	$(z_{II} - z_I)$ (m)

Another useful method for calculating head losses in fittings was proposed by William B. Hooper in *Chemical Engineering.* See Section 3.6.

The flow losses in both suction and discharge systems must be evaluated. For variable flow systems a Q-h_f curve should be produced for both systems; the suction system curve should also show NPSHa/NPIPa:

$$\frac{\text{Net Positive Suction Head Available}}{\text{Net Positive Inlet Pressure Available}}$$

Both functions are the liquid pressure minus the vapour pressure. It is best for the system designer to use the top of the pump foundations as the datum level; the pump manufacturer can easily adjust the data to satisfy the pump configuration. Centrifugal pumps use the centre of the impeller eye as the datum whereas reciprocating pumps use the centre of the suction connection.

In long systems, or systems where the flow velocity is high; these terms are of necessity vague; the possibility of surge or water-hammer due to transients or fast acting valves must be considered. Very high peak pressures can be generated and the system/pump must be either designed to cope or protected from these effects. Relief valves and bursting discs can be fitted for protection, standard relief valves can be damaged in the event. Pulsation dampers and air vessels may be fitted to reduce the magnitude of the pressure pulses generated. See Chapter 5 for more information.

Some positive displacement pumps produce uneven flows. Reciprocating pumps, due to the speed variation of the pistons or plungers, produce a cyclic flow variation in the suction and discharge systems. The flow variation in the suction system produces addition pressure losses which must be considered. The Hydraulic Institute (USA) and VDMA (Germany) have both produced data to allow this phenomenon to be evaluated.

In general, it is better to have more NPSHa/NPIPa than less. A wider range of pump types will be available if there is plenty NPSHa/NPIPa. Low NPSHa/NPIPa usually means bigger pumps running slowly.

Conversely high suction pressure can also cause problems; increased thrust, bearing cooling problems and reduced efficiency. The suction pressure should be considered relative to the discharge pressure. If the discharge pressure is greater than 16 barg and the suction pressure is greater than 20% of the discharge, it is advisable to consult manufacturers about possible problems and cost/size implications.

NOTE: The treatment of pipe flow losses in Section 3.2 applies to Newtonian liquids of which the commonest is water. Non-Newtonian liquids require special attention regarding the determination of pressure losses. A special type of flow loss is caused by control valves in the pipe system. The control valve pressure losses are dealt with from a sizing point of view in Chapter 5 and from the regulation viewpoint in Chapter 6.

3.3 Liquid-solid mixtures

3.3.1 General

Liquid-solid mixtures, solid particles in liquid, behave as non-Newtonian liquids. There are many parameters which affect the flow characteristics of liquid-solid mixtures. All pumping of solids, i.e. transportation of solid materials in liquid, is characterised by a certain degree of uncertainty. It is normally recommended that testing is carried out, with representative solids, to evaluate critical velocity, head loss and wear characteristic before the design is finalised.

Some of the main aspects of the behaviour of liquid-solid mixtures are set out in the following Sections, along with guideline values for the estimation of the more important parameters for certain liquid-solid mixtures.

3.3.2 Homogeneous mixtures

When the solid particles distribute themselves evenly throughout the cross-section of the pipe, without any concentration gradients, we speak of a homogeneous mixture or suspension. In practice, homogeneous mixtures are considered to occur when the particle size is less than about 50μm (0.050 mm).

Homogeneous mixtures with sufficiently high particle content behave as time-independent, plastic, non-Newtonian liquids. These kinds of liquid to some degree follow Binghams Law, see Chapter 2, Section 2.4 and also Figure 3.13.

The pressure drop for laminar flow in straight piping measured in metres head of mixture is:

$$h_f = \frac{32}{\rho g} \cdot \frac{l}{d} \cdot \left(\frac{t_0}{6} + \frac{\tan\gamma \cdot v}{d}\right) \quad \text{Equ 3.25}$$

where:

h_f	=	loss of head (m)
ρ	=	density of liquid-solid mixture (kg/m^3)
g	=	acceleration due to gravity (m/s^2)
l	=	length of pipe (m)
d	=	diameter of pipe (m)
t_0	=	viscous shear stress at boundary (N/m^2)
$\tan\gamma$	=	plastic dynamic viscosity (Ns/m^2)
v	=	velocity of flow (m/s)

This is illustrated in Figure 3.14 for a homogeneous liquid- solid mixture (ρ_p = 2600 kg/m^3, d = 50 mm)

The viscous shear stress at boundary τ_0 and plastic viscosity, $\tan\gamma$, can be determined with the aid of a viscometer or by mea-

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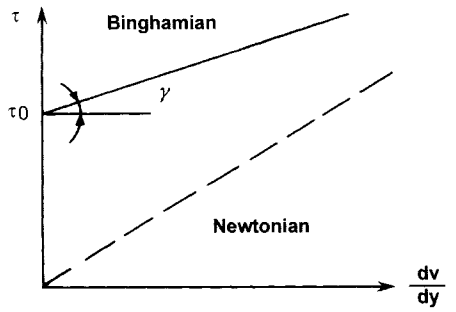

Figure 3.13 Homogenous suspensions with flow characteristics as per Binghamian and Newtonian behaviour

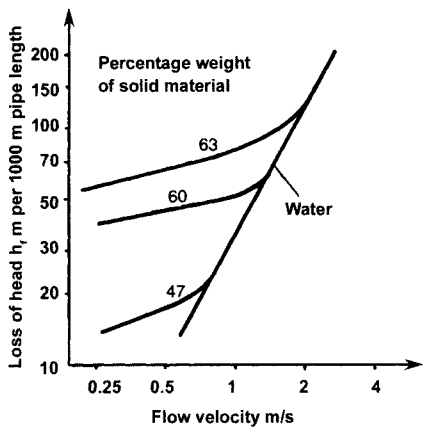

Figure 3.14 Examples of loss of head in pipe flow

suring the pressure drop in a test loop. Plastic viscosity, tan γ, is also called apparent viscosity in some texts.

The loss of head for turbulent flow in straight pipe, measured in metres of mixture is:

$$h_f = l \cdot \frac{l}{d} \cdot \frac{v^2}{2g} \qquad \text{Equ 3.26}$$

i.e. exactly the same formula as for Newtonian liquids. The loss coefficient must be determined using the apparent viscosity.

Expressed in metres liquid column, the loss of head for a homogeneous liquid-solid mixture is the same as for pure water in turbulent flow.

The transition from laminar to turbulent flow occurs, as for water, at Re ≅ 2300 but with the difference that the apparent viscosity should now be used for Reynolds Number. The velocity of flow at transition V_{cr} for Re = 2300 is:

$$V_{cr} \cong \frac{1150 \cdot \tan\gamma}{r \cdot d}\left(1 + \sqrt{\frac{1 + \tau_0 \cdot r \cdot d^2}{3600 \cdot \tan^2 g}}\right) \qquad \text{Equ 3.27}$$

where:

V_{cr}	=	critical velocity of flow at transition (m/s)
tan γ	=	plastic viscosity (Ns/m²)
ρ	=	density of liquid-solid mixture (kg/m³)
t_0	=	shear stress at boundary (N/m²)
d	=	diameter of pipe (m)

Sedimentation in the pipe is avoided completely if the flow velocity in the pipe is greater than V_{cr}. Thus V_{cr} is of considerable importance when sizing pump installations for homogeneous mixtures in pipes. In short or carefully monitored pipes, $v < V_{cr}$ can sometimes be accepted.

There is an optimum solids concentration for the transportation of solid material by hydraulic methods (see Figure 3.15). If the concentration is too low, the liquid-solid mixture loses its plastic

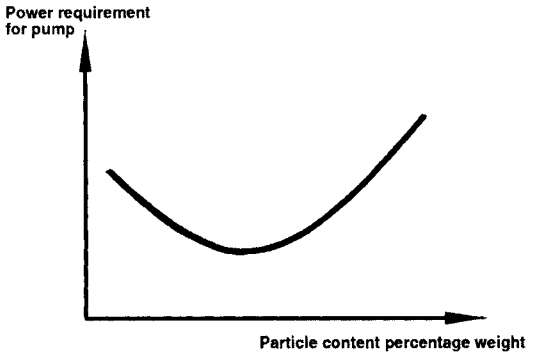

Figure 3.15 Optimum particle content for homogeneous liquid-solid mixtures for the transportation of a given quantity of solid material per unit time

characteristic and with it the possibility of making use of the advantageous "plug" flow. Besides which an unnecessarily large amount of liquid is pumped per kilogram of solid material if the liquid is discarded. If the particle content is too high the transportation must be carried out at increased velocity due to increased transitional velocity, V_{vc}, so as to avoid sedimentation.

If, for example, the particle concentration in the liquid-solid mixture in Figure 3.14 were to increase from 60% to 63%, the velocity of flow would have to increase from about 1.7 to 2.2 m/s. Thereby increasing the transportation capacity by a factor of 1.4, the loss of head by about 1.7 and the pump power consumption, with pipe losses only, by a factor of about 2.3. When the transportation capacity is constant but using a smaller pipe diameter the power requirement of the pump increases 1.7 times. The particle content is thus a critical economic parameter.

3.3.3 Heterogeneous mixtures

Heterogeneous mixtures or suspensions, that is to say liquid-solid mixtures with concentration gradients across the pipe section, are considered in practice to occur for particle sizes of more than about 50 μm (0.050 mm). The main difference compared with homogeneous liquid-solid mixtures is that the liquid and the particles in a heterogeneous mixture retain their identity. The flow is a two - phase flow. The viscosity of a heterogeneous mixture in principle is the same as that for the pure liquid.

The solid particles are conveyed suspended in the liquid if the particles are small and the velocity high, or by jumps if the particles are large or the speed low, as shown in Table 3.3..

Particle size mm	Normal means of transportation
0.05 to 0.2	suspended
0.20 to 1.5	combined suspended
> 1.50	jumps

Table 3.3 Normal means of transportation for various particle sizes

For all practical purposes, particle sizes in the order of 0.05 to 0.2 mm are transported suspended in the liquid. Liquid-solid mixtures with particle sizes down towards the lower limit, 0.05 to 0.1 mm, comprise the boundary with the homogeneous mixtures. The most effective way of minimizing the risk of sedimentation and blockage of the pipe is to maintain the pipe flow in the turbulent regime. The pressure losses, expressed as loss of head in metres mixture, can then be calculated in the same way as for the liquid.

Liquid-solid mixtures of particles with sizes up towards the upper limit, 0.1 to 0.2 mm, are on the boundary with the next group where the combined means of transportation prevails. The particle region 0.05 to 0.2 mm is not so well understood as other size ranges.

Mixtures with particles of 0.2 mm have been the subject of comprehensive pressure loss measurements. (See work by Durand, Condolios et al., see Section 3.6.) Figure 3.16 shows an example of a loss diagram for a horizontal pipe.

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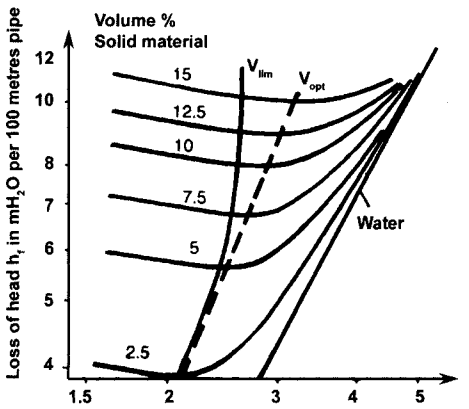

Figure 3.16 Example of loss of head in heterogeneous suspension, sand ρ_p - 2650 kg/m^3, d_p mean ≅ 0.4 mm, d = 200 mm

The experimental results shown cover the region:

d	=	pipe diameter - 0 to 600 mm
d_p	=	particle size - 0.2 to 25 mm
K_{vol}	=	volumetric concentration - 2.5 to 15%
ρ_p	=	particle density liquid 0 to 5000 kg/m^3 water

The loss of head expressed in metres of water (m H_2O) is greater than for pure water and exhibits a minimum at the optimum flow velocity v_{opt}. The loss of head for liquid-solid mixtures of d_p > 0.2 mm flowing in horizontal pipes can be assessed using the formula:

$$h_f = \lambda \cdot \frac{l}{d} \cdot \frac{v^2}{2g} (1 + k_f) \qquad \text{Equ 3.28}$$

where:

h_f	=	loss of head (m H_2O)
λ	=	loss coefficient for water
d	=	pipe diameter (m)
v	=	velocity of flow (m/s)
k_f	=	supplementary loss factor over and above that of pure water

The supplementary losses above those for pure water can be assessed from:

$$k_f = 85 \cdot K_{vol} \left[\frac{g \cdot d}{v^2} \cdot \left(\frac{\rho_p}{\rho_v} - 1 \right) \cdot k_p \right]^{3/2} \qquad \text{Equ 3.29}$$

where:

K_{vol}	=	proportion of solid material by volume
ρ_p	=	particle density (kg/m^3)
ρ_v	=	liquid density (kg/m^3)
k_p	=	factor for particle characteristics

The particle factor is assessable from:

$$k_p = \frac{V_s}{\sqrt{\frac{4}{3} \cdot \left(\frac{\rho_p}{\rho_v} - 1 \right) \cdot g \cdot d_p}} \qquad \text{Equ 3.30}$$

where:

v_s	=	sedimentation velocity for a single particle in still water (m/s)
d_p	=	particle diameter (m)

Some text books use the dimensionless "particle resistance coefficient" C_D instead of the particle factor k_p. The relationship between them is given by:

$$k_p = \frac{1}{\sqrt{C_D}} \qquad \text{Equ 3.31}$$

The optimum flow velocity v_{opt}, i.e. the flow velocity for minimum loss of head at any given volumetric concentration, can be assessed from equation 3.32.

$$v_{opt} = 3.5 \cdot K_{vol}^{0.333} \left[g \cdot d \cdot \left(\frac{\rho_p}{\rho_v} - 1 \right) \cdot k_p \right]^{1/2} \qquad \text{Equ 3.32}$$

For particles >0.2 mm and density ρ_p = 2650 kg/m^3, some text books give:

$$v_{opt} = 2.9 \cdot K_{vol}^{0.15} \cdot \sqrt{g \cdot d} \qquad \text{Equ 3.33}$$

Another velocity of interest is the "limiting" velocity, below which the particles are in continuous contact with the bottom of the pipe. The limiting velocity can be evaluated, at testing, from:

$$v_{lim} = F_{lim} \sqrt{2g \cdot d \left(\frac{\rho_p}{\rho_v} - 1 \right)} \qquad \text{Equ 3.34}$$

where the factor F_{lim} depends upon the volumetric concentration and size of particles and is determined experimentally. See Figure 3.17.

Liquid-solid mixtures with particle contents of different sizes are dealt with as follows:

- Small particles which form a liquid-solid mixture with the pure liquid at the velocity used are considered to be a part of the carrier liquid. The liquid and the small particles (d_p less than perhaps 0.1 mm, certainly less than 0.05 mm) constitute a new liquid with high density and viscosity values.
- The larger particles together with the carrier liquid form a heterogeneous mixture. Where there is a variation of size within the group of larger particles, a weighted mean value is used to determine the particle factor k_p.

Small particles increase the density of the carrier liquid and reduce the ratio of densities of transported particles and carrier liquid. Because of this both the supplementary losses and the limiting velocity for minimum loss and sedimentation are reduced. Additions of small particles can thus be economically attractive in the transportation of larger particles.

The sedimentation velocity v_s used to calculate the particle factor k_p should be quantified experimentally. The effect of the density of the liquid and its viscosity, together with the geometrical shape of the particle all affect the particle factor. The measured sedimentation velocity for sand in water is illustrated in Figure 3.18. This is in accordance with Richards for quartz sand (ρ_p = 2650kg/m^3) in water (10°C, ν = 1.3 x 10^{-6} m^2/s) compared

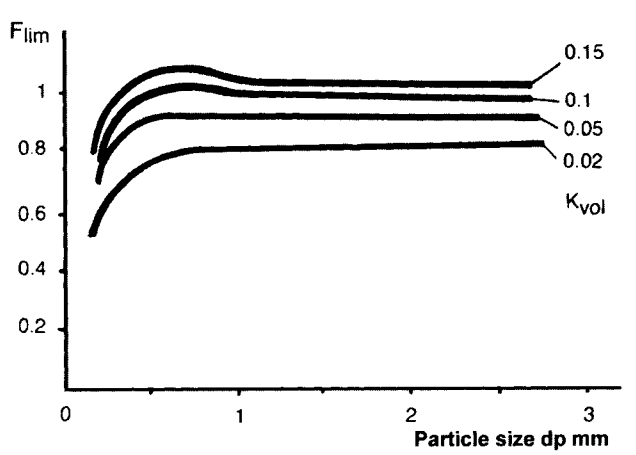

Figure 3.17 Factor F_{lim} for the determination of the limiting velocity V_{lim}

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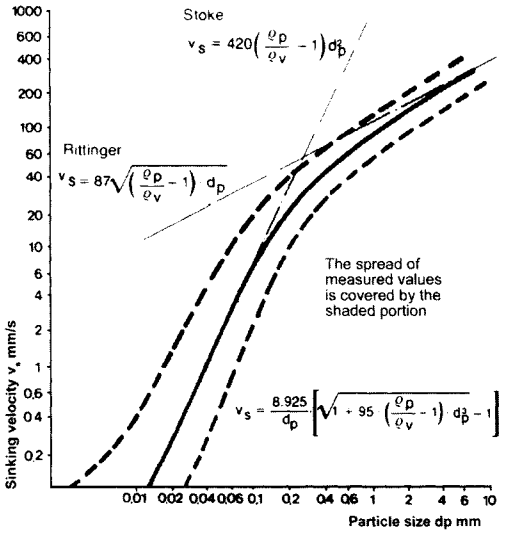

Figure 3.18 Measured sedimentation velocity

with various theoretical expressions

It is extremely difficult to give general guideline values for the most suitable flow velocity for the hydraulic transportation of solid materials. The values shown in Table 3.4 are recommended.

Solids	Mean velocity m/s
waste from ore dressing plant	1.5 - 2.1
dust from precipitators	1.5 - 1.8
fly ash	1.8 - 2.1
fine sand	2.4 - 3.0
normal sand with fine	3.4
coarse sand without fine	3.7 - 4.0
shingle, max. 13 mm	4.3

Table 3.4 Flow velocity for hydraulic transportation of solid materials

These values are based on pipe diameters between 100 and 200 mm. Note the effect of the pipe diameter on the limiting velocity in equation 3.34.

3.3.3.1 Special note

The formulae presented for the evaluation of properties for liquid-solid mixtures are the result of extensive research. Research is always being conducted and more accurate expressions may be available. However all published data is approximate. The equations are derived by finding the "best fit" to test results. It is very unlikely that any of the test results would closely approximate a particular actual operating condition and there may be no way of evaluating how far the "best fit" equation is from the closest suitable test results.

In practice, it will be the size distribution/content of the solids which determines the flow properties. When solids such as copper/iron ore, coal are crushed, variable quantities and sizes of fines are also produced. Model tests on representative crushed samples are essential to confirm calculated results. Wear tests, such as Miller Number, ASTM G75, should also be undertaken to assess what precautions, if any, are required against wear.

Discussions with pump manufacturers of several pump types, should be held as soon as approximate figures are available for solid sizes and distribution, flows and pressures. The discussions should focus on:

- **Efficiency** — how efficient will the pump be and how will the efficiency be affected by changing operating conditions and wear,
- **Wear** — which components will wear, predicted life of components and accessibility for maintenance,
- **Adjustment** — how can the pump be adjusted, at site, to optimize efficiency and component life for changing operating conditions. Multi-stage centrifugal pumps can be de-staged or have impeller diameters modified; reciprocating pumps can have different diameter pistons or plungers fitted,
- **NPSHr** — some pumps require a lot of NPSHa and this can radically effect the design of suction systems.

3.3.4 Paper pulp

3.3.4.1 General

The behaviour of pulp suspensions when flowing through a pipe is unusual from several aspects. The loss of head for chemical pulps goes through a maximum and a minimum value when the flow velocity increases from zero, and the head loss is finally less than the value for pure water.

Referring to Figure 3.19, for velocities up to point D, after the intersection of the two curves, the suspension flows like a plug surrounded by a thin boundary layer in which the whole of the viscous flow occurs. The character of the boundary layer varies according to the position of the working point on the curve:

- A to B — The surface of the plug is disturbed by intermittent contact with the pipe wall.
- B to C — The boundary layer consists mainly of pure water in laminar flow.
- C to E — The flow in the boundary layer is turbulent.

Plug flow can be regarded as ended when the stress at the edge of the plug is equal to the ultimate shear stress of the network of fibres which form the plug. When the velocity increases beyond this point the plug breaks up and finally the whole cross section flows turbulently.

The friction loss curves for mechanical pulps do not usually have any maximum or minimum. The flow characteristics are modified by the relatively low strength of the network and the high proportion of non-fibrous solid material.

3.3.4.2 Calculation of pipe flow losses

Pipe flow losses can be determined with the help of a number of diagrams shown in Figures 3.20 and 3.21.

The non-dimensional flow parameter S is defined as:

$$S = \left(\frac{V^5 \cdot \rho^2 \cdot h}{\tau_D^3} \cdot d\right)^{1/6} \qquad \text{Equ 3.35}$$

and similarly the non-dimensional loss parameter F:

$$F = \frac{r \cdot g \cdot h_f \cdot d}{4 \cdot \tau_D \cdot l} \qquad \text{Equ 3.36}$$

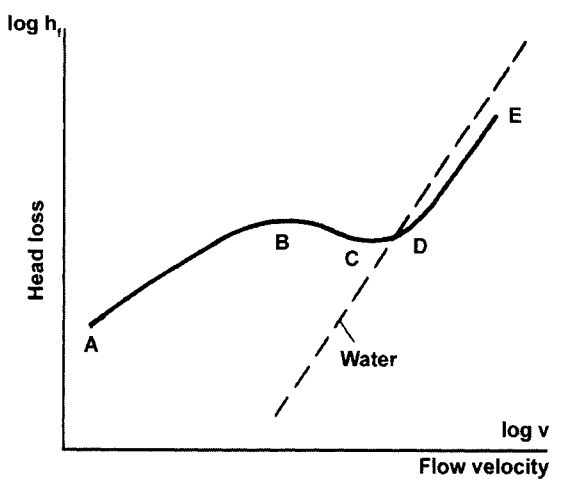

Figure 3.19 Loss of head for chemical pulps
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The shear stress τ_D is a measure of the strength of the plug of plaited fibres which occurs at most flow velocities in practical operations.

$$\tau_D = \tau_D^1 \cdot f_F \cdot f_R \cdot f_M \cdot f_D \qquad \text{Equ 3.37}$$

where:

h_f	=	loss of head (m)
l	=	pipe length (m)
v	=	flow-velocity (m/s)
k	=	surface roughness of pipe (m)
d	=	pipe diameter (m)
ρ	=	density of suspension (kg/m^3)
μ	=	dynamic viscosity (Ns/m^2)
τ	=	shear stress (N/m^2)
F_k	=	length/diameter ratio of fibre

If the pulp never dries out, $f_D = 1$. If it has dried out and is absorbed again, $f_D = 0.75$.

The diagrams, especially Figure 3.20, apply to chemical pulps. The different characteristics of mechanical pulps shown in Figure 3.21 (fibres, networks, etc.) give the loss curve a different shape and generally increased losses. Correction factors should therefore be used. Approximate values can be obtained, however, using the above method provided that:

- the loss parameter F never assumes a value lower than the maximum point in Figure 3.20 for the section of the curve S >0.6.
- $F_k = 1$
- a safety factor of 1.2 should be used.

The guideline values for head losses in bends and fittings are given in Section 3.4.4 in the form of equivalent pipe lengths.

It is typical of pulp suspensions, as for all plastic non-Newtonian liquids, that a certain yield shear stress (stiction) has to be overcome before a flow can be made to start up at all. This stiction can be reduced considerably by dilution.

3.3.4.3 Calculation example

A chemical pulp, 281 m^3/h, has to be pumped through a 70 m long stainless steel pipe diameter 300 mm. The pulp has never been dried and has a fineness grading 600 C.s.f., the temperature is 30°C and the concentration 2.7% absolutely dry. The average fibre length is 2.4 mm and mean diameter 40 μm. Determine the loss of head through the pipe.

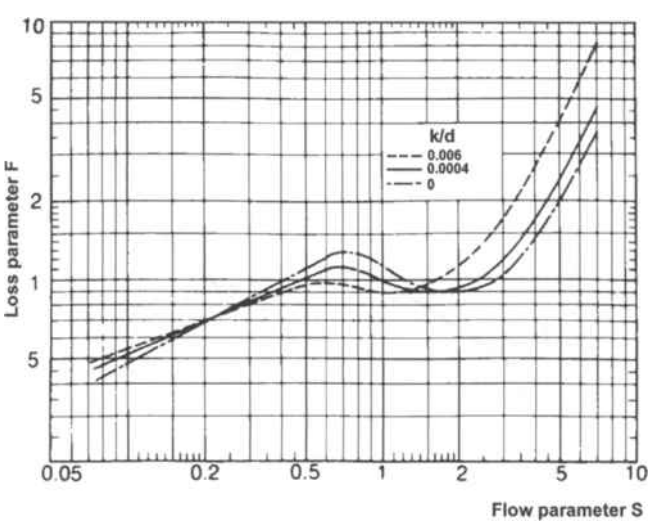

Figure 3.20 Pipe flow characteristics of chemical pulp
Courtesy of ABS Group

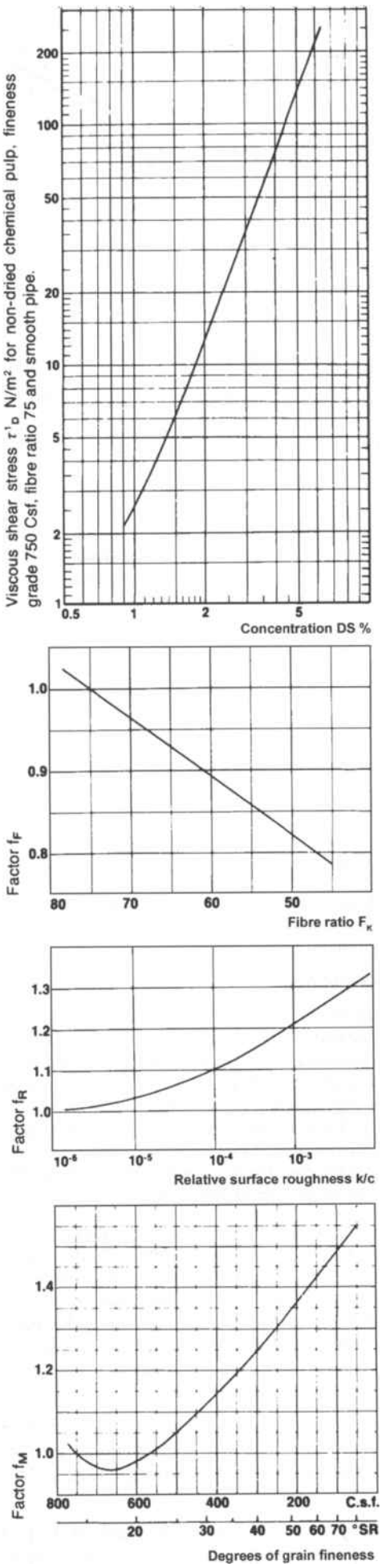

Figure 3.21 Properties of chemical pulps
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Fibre ratio $F_K = 2.4/0.040 = 60 \rightarrow f_F = 0.89$

Stainless steel pipe k/d = 0.045/300 = 1.5 x 10-4 $\rightarrow f_R = 1.11$

Fineness grading 600 Csf $\rightarrow f_M = 0.98$

Undried chemical pulp $\rightarrow f_D = 1$

DS content 2.7 % $\rightarrow \tau^1{}_D = 27$ N/m^2

where:

DS = proportion by weight of fibre (absolutely dry) (%)

Shear stress $\tau_D \rightarrow 27 \cdot 0.89 \cdot 1.11 \cdot 0.98 \cdot 1 = 26.1$ N/m^2

Water at 30°C $\rightarrow \mu = 0.8 \cdot 10^{-3}$ Ns/m^2, $\rho \sim 1000$ kg/m^3

Flow velocity

$$v = \frac{Q}{\frac{\pi \cdot d^2}{4}} = \frac{281 \cdot 4}{3600 \cdot \pi \cdot 0.3^2} = 1.1 \text{m/s}$$

The flow parameter

$$S = \frac{1.1^5 \cdot 1000^2 \cdot 0.8 \cdot 10^{-3}}{26.1^3 \cdot 0.300} = 0.79$$

Read off loss parameter F = 1.05 (Figure 3.20)

$$h_f = \frac{4 \cdot \tau_D \cdot l}{\rho \cdot g \cdot d} \cdot F = \frac{4 \cdot 26 \cdot 1.70}{100 \cdot 9.81 \cdot 0.3} \cdot 1.05$$

$$h_f = 2.4 \text{m}$$

3.4 Pressure losses - nomograms and diagrams

3.4.1 General

Pre-computed pressure losses in pipes for a number of commonly applied cases of flow are set out in this section in the form of diagrams. These diagrams and nomograms can be used to calculate the losses in a system, i.e. flow losses in the pipes, valves, bends, etc. This method is quick and provides acceptable results if carried out accurately.

A number of diagrams for the determination of the associated values of velocity, frictional losses, diameter and volume flow are presented. There are also diagrams for determining the flow velocity in pipes in relation to the volume flow, mass flow and diameter together with diagrams for the determination of head loss (m) in bends fittings etc., related to the velocity of flow.

It is most practical when sizing pipes to be able, right from the outset, to assess the approximate economic flow velocity. By this means, the final economic sizing can be carried out rapidly, since fewer alternatives have to be tested. Table 3.5 gives guidelines for normal flow velocities in some common cases.

Liquid application	Flow velocity m/s
Dirty water	
Mine water lines	1.75 to 2.5
Hot water	
Condensate	1.0 to 1.5
District heating distribution	1.0 to 2.5
District heating service main	0.75 to 2.0
Domestic heating circulation	0.3 to 0.6
Feed water lines	1.2 to 4.3
Crude oil	
Oil pipelines	0.25 to 1.8
Fuel oil	
Truck loading	5 to 6
Lube oil	
Drain line (1)	0.15 to 0.75
Feed lines	1.0 to 2.3
Pump discharge line	0.6 to 1.8
Pump suction line (2)	0.45 to 1.5
Pump suction line (3)	0.3 to 1.0
Petrol	
Pump discharge	1.0 to 1.4
Pump suction	0.45 to 0.75
Refrigerant	
Condenser to receiver	≤ 1.0
Receiver to system (4)	≤ 1.5
Sewage	
Pump discharge, centrif	≥ 2.4
Pump suction, centrif	≥ 1.5
Warm water	
Economiser tubes	0.75 to 1.5
Water	
Pump discharge, centrif	8 to 15
Pump discharge, recip	3.0 to 4.5
Concrete suction bays	≤ 0.3
Pump suction headers	≤ 1.0
Pump suction, centrif	0.45 to 1.5
Pump suction, recip	≤ 0.6
Consumer supply	1.0 to 1.4
Service mains	0.6 to 1.5
Trunk main	1.0 to 3.0

(1) 1:50 fall minimum, half full, (2) flooded suction, (3) suction lift, (4) 0.115 bar / 25m maximum

Table 3.5 Guideline flow velocities for general applications

High flow velocities give small pipe dimensions giving low plant procurement costs; but cause greater pump power consumption through greater frictional losses resulting in high running costs.

Quite apart from economics, other factors can affect the choice of flow velocity:

- Noise in central heating equipment
- Erosion in copper hot water pipes
- Static electricity for liquids with low flash points
- Risk of cavitation in suction lines
- Risk of cavitation with sudden changes in velocity
- Sedimentation with liquid-solid mixtures, i.e. critical velocity
- Vibration of pipework
- Surge and pressure pulsation problems

If the NPSHa/NPIPa margin is small special attention should be given to pump suction systems. The best systems are short, straight and one pipe size larger, at least, than the pump suction connection. Any bends should be more than five pipe diameters from the pump connection. If the NPSH/NPIP margin is small it is critically important to evaluate all system losses for all flow rates. If the suction system causes pump cavitation, operational problems will reduce the maintenance interval. Typical problems include:

- Loss of pump capacity
- High rotating/reciprocating parts wear
- Sealing problems
- Fatigue failures

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- Discharge pipework noise and vibration
- Pipe connection leaks

It should be remembered that thermodynamic data is for a pure substance with no contaminants. Fresh water will contain dissolved gases. Gases will evolve from solution at a much higher pressure than the vapour pressure of the water; typically at 8m absolute when cold. Gas evolution causes the same problems, and looks exactly the same, as cavitation. For liquids with dissolved gases, the NPSHa/NPIPa margin must be based on the gas evolution pressure, not liquid vapour pressure. A pump cannot distinguish between released gas and saturated vapour, the effects and the damage are the same.

Thermodynamic data must be comprehensive if used for pump applications. Data on compressed liquid state must be available as well as saturation line data.

The higher velocities listed in Table 3.5 demand better pipework designs and high quality manufacture and assembly. Poorly fitted slip-on flanges, badly aligned, will cause extra turbulence and losses. Bent pipe is preferable to fabricated bends. In most cases, a large radius should be used in place of a short one. The minimum of weld neck flanges, or connectors which use locating seal rings, such as Grayloc's, will preserve good flow patterns.

Losses in bends and valves etc. can be expressed as equivalent straight pipe lengths by using the values in Table 3.6. The values for valves are mean values and vary according to the design. By adding the equivalent length of pipe and the straight length, the total pipe length is obtained enabling the flow losses, expressed as head (m), to be calculated from the diagrams.

As pointed out in Section 3.2.3, the equivalent pipe length depends upon the straight pipe loss coefficient A, i.e. pipe roughness, Reynolds Number etc. The numerical values in Table 3.6, therefore must be regarded as approximate guideline values.

Obstacle type	Equivalent pipe length	
	d = 10 - 50 mm	d = 80 - 400 mm
Gate valve	15 - 10 x d	10 - 5 x d
Check valve (with flap)	200 - 150 x d	150 - 100 x d
Angle valve (seating)	400 - 200 x d	150 - 100 x d
Stop valve (straight seating)	1000 - 500 x d	500 - 400 x d
Pipe bend 45°	10 x d	10 x d
Pipe bend 90°	30 - 20 x d	15 - 10 x d
Tank outflow (rounded)	10 - 5 x d	10 - 5 x d
Tank outflow (sharp edged)	40 - 30x d	40 - 30 x d
T-branch through side outflow	50 - 40 x d	40 x d

Table 3.6 Conversion factors for equivalent pipe lengths

Alternatively, losses in bends, valves etc., can be separately calculated by converting the values for ζ obtained from Figure 3.11 into metres head (column of liquid) at specific flow velocities with the help of the diagrams. Note that the stated value of ζ is for turbulent flow. In the case of laminar flow conversion to equivalent pipe length is the most convenient way to calculate these losses. Figure 3.22 shows the flow velocity in pipes for volume flow (m^3h and l/min), and pipe diameter (m). Figure 3.23 shows the losses of head in bends and fittings

The applicability of the diagrams is wider than indicated. Figures 3.24 to 3.41 give frictional loss diagrams for calculating pressure losses in a wide range of pipelines. The diagrams for water can be used for all water-like Newtonian liquids, i.e. for liquids with corresponding viscosity. Note, however, that if the pressure loss is stated in Pa (Pascal) the density must also coincide with that of water.

The diagrams for oils can be used for all Newtonian liquids with higher viscosities than water, under the same conditions.

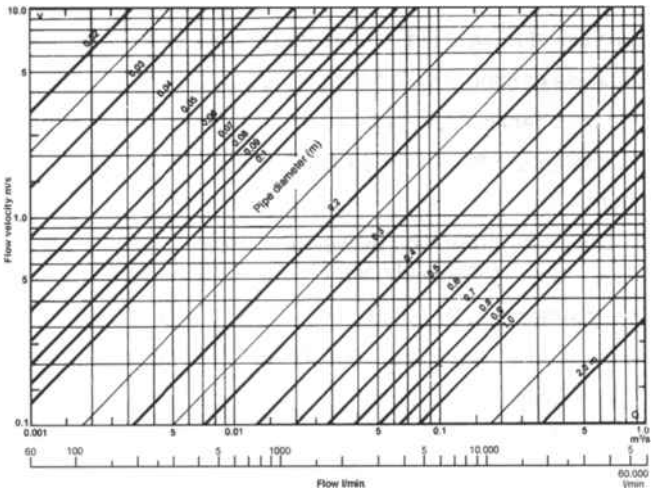

Figure 3.22 Flow velocity in pipes in for volume flow and diameter. (Interpolate between the lines for other diameters)

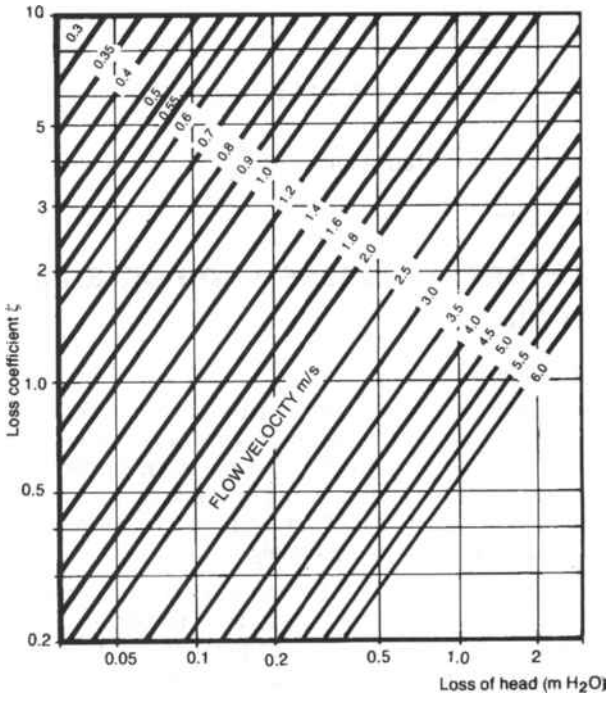

Figure 3.23 Relation between loss coefficient and loss of head in bends and fittings

3.4.1.1 Calculation example

Calculate the total loss of head in 3 x 90° short radius pipe bends in a 100 mm diameter pipe for a water flow of 700 l/m.

From Figure 3.22, velocity of flow = 1.5 m/s

From Figure 3.11, $\zeta = 3 \times 0.4 = 1.2$

From Figure 3.23, loss of head = 0.14 m

3.4.2 Water

The diagrams also apply to other Newtonian liquids having similar viscosity to that of water when the pressure loss is expressed as head (metres of liquid column).

3.4.2.1 Calculation example

A flow of 10 l/s is required through a 120 mm diameter pipe, 200 m long. Calculate the flow velocity and the loss of head. From the diagram, the velocity is 0.9 m/s and loss of head 200 x 7/1000 = 1.4 m

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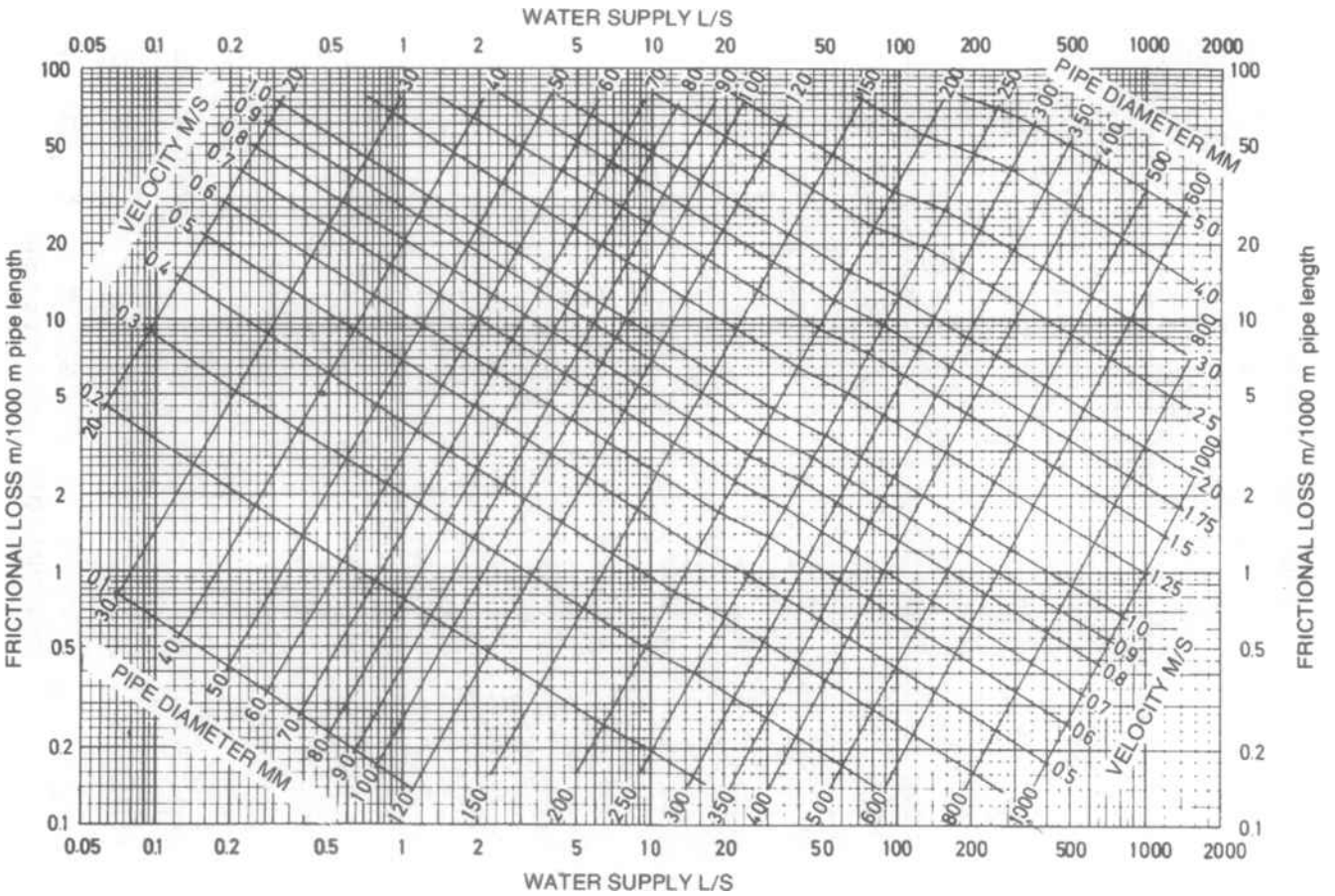

Figure 3.24 Frictional loss diagram for water at +10°C, k = 0.01 mm for diameter ≤ 200 mm, k = 0.05 mm for diameter > 200 mm. Pipe type: steel, Cu, PE, PVC

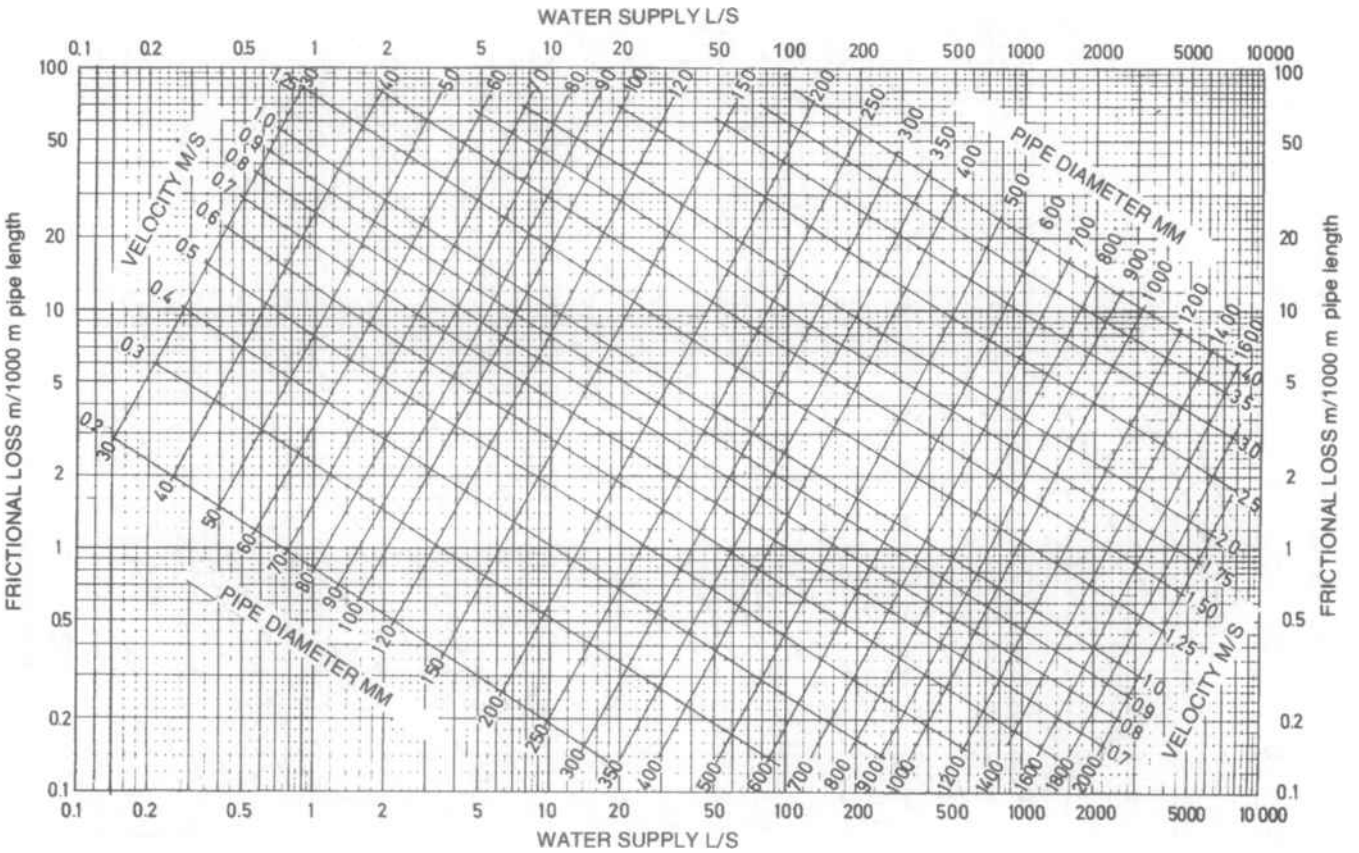

Figure 3.25 Frictional loss diagram for water at +10 ° C, k = 0.1 mm. Pipe type: steel, cast iron, GRP

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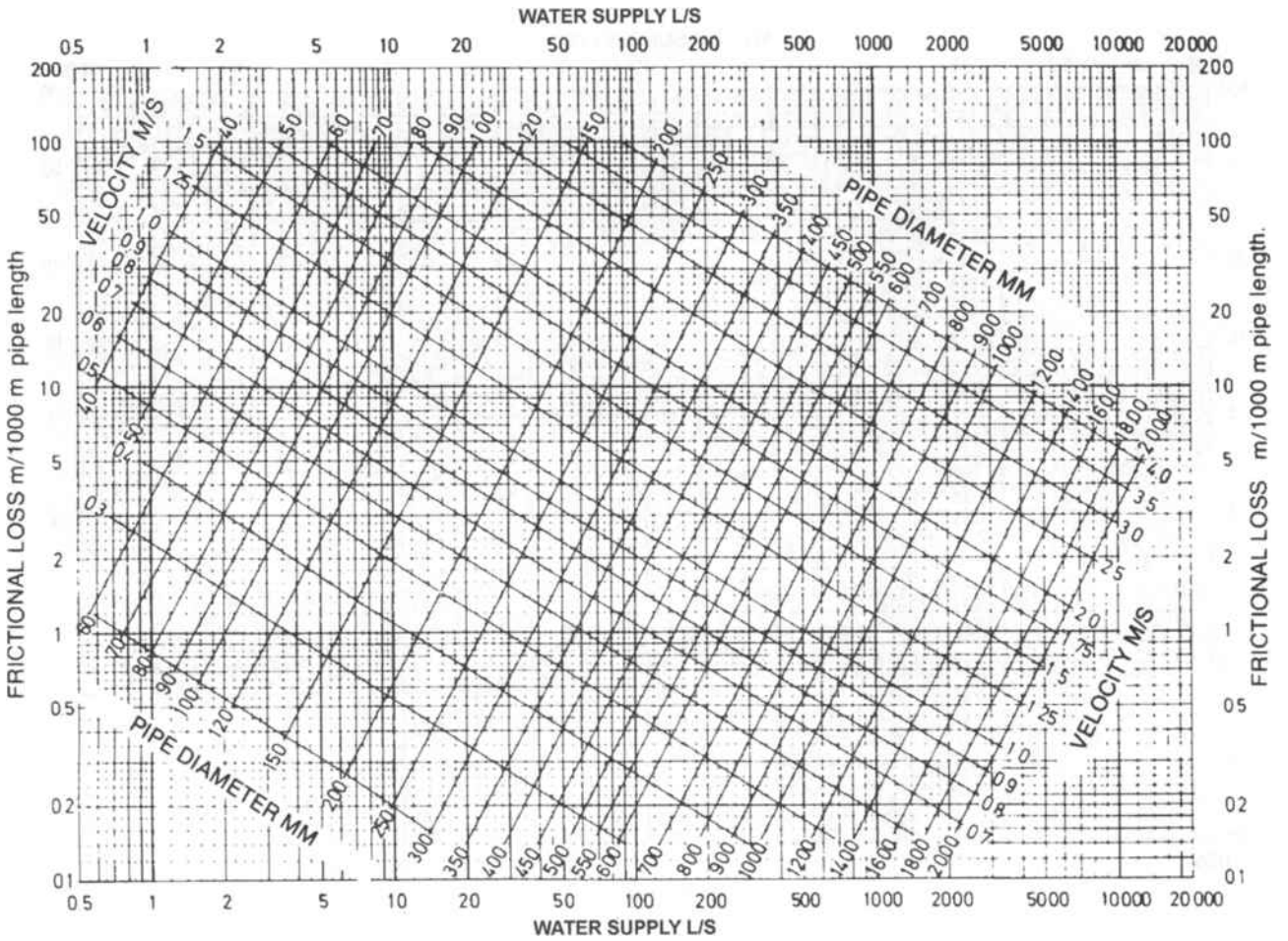

Figure 3.26 Frictional loss diagram for water at + 10°C, k = 0.2 mm. Pipe type: steel, cast iron, concrete, wood

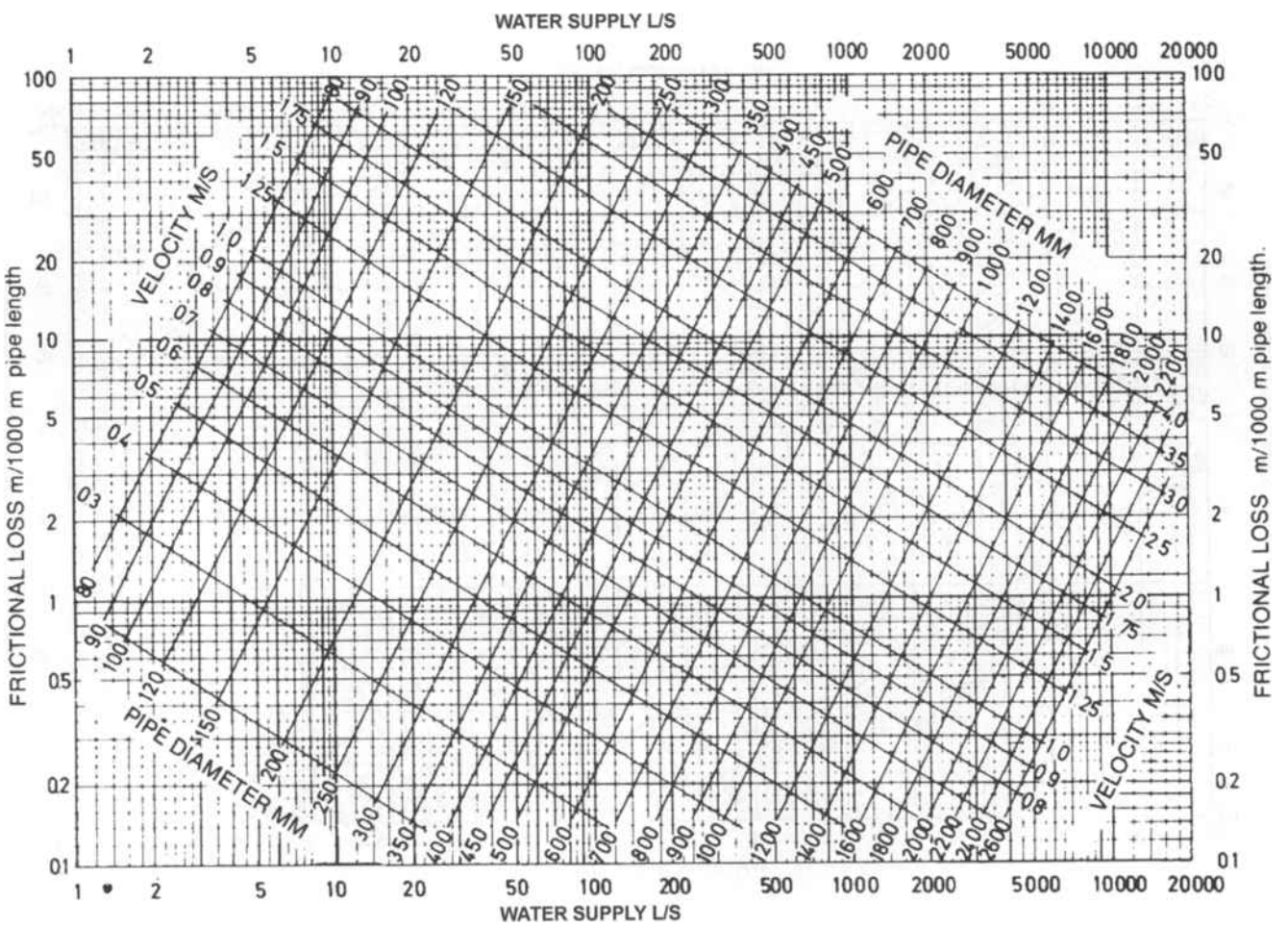

Figure 3.27 Frictional loss diagram for water at +10°C, k = 0.5 mm. Pipe type: steel, cast iron, concrete

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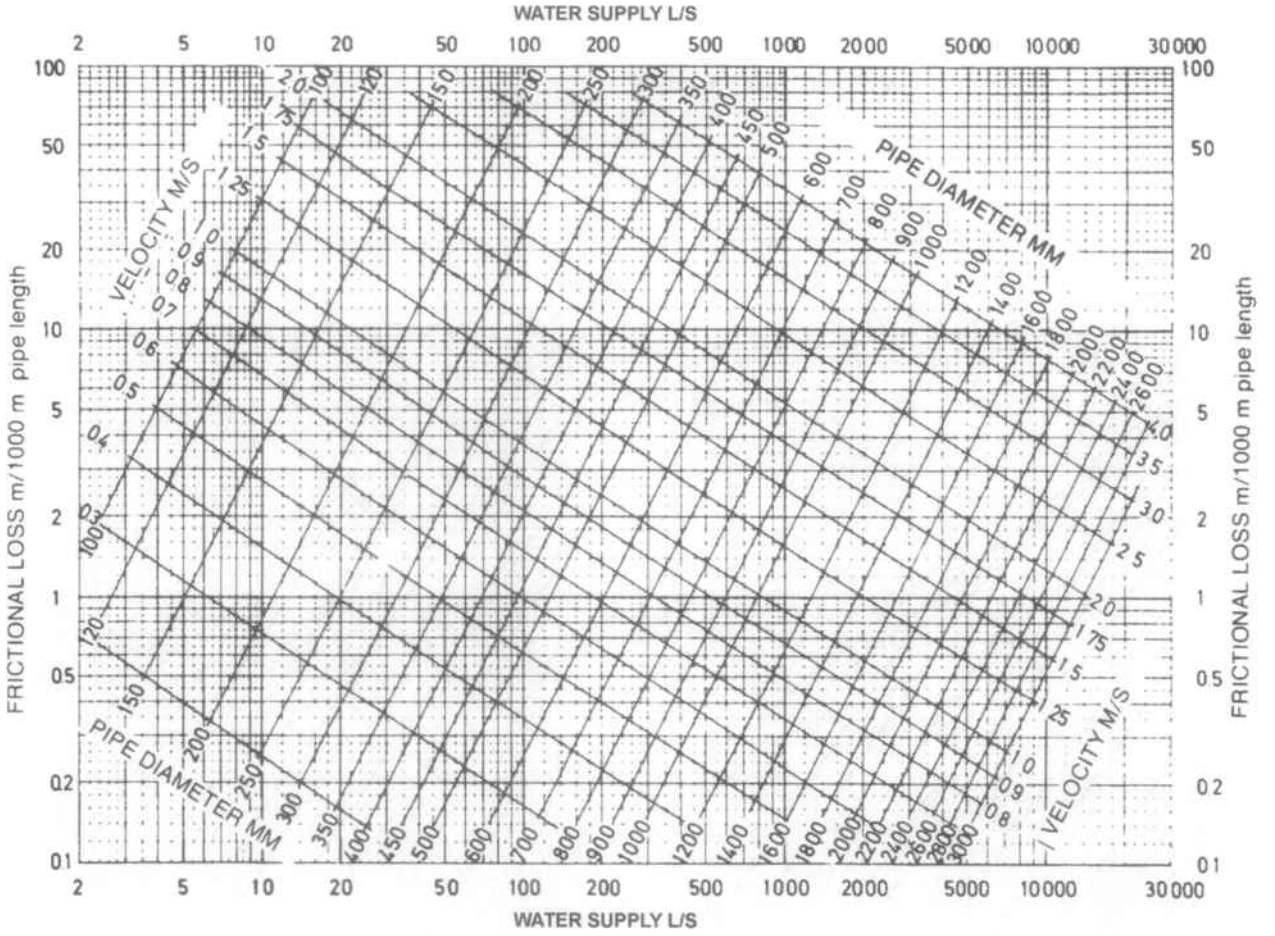

Figure 3.28 Frictional loss diagram for water +10°C, k = 1.0 mm. Pipe type: cast iron, concrete, earthenware

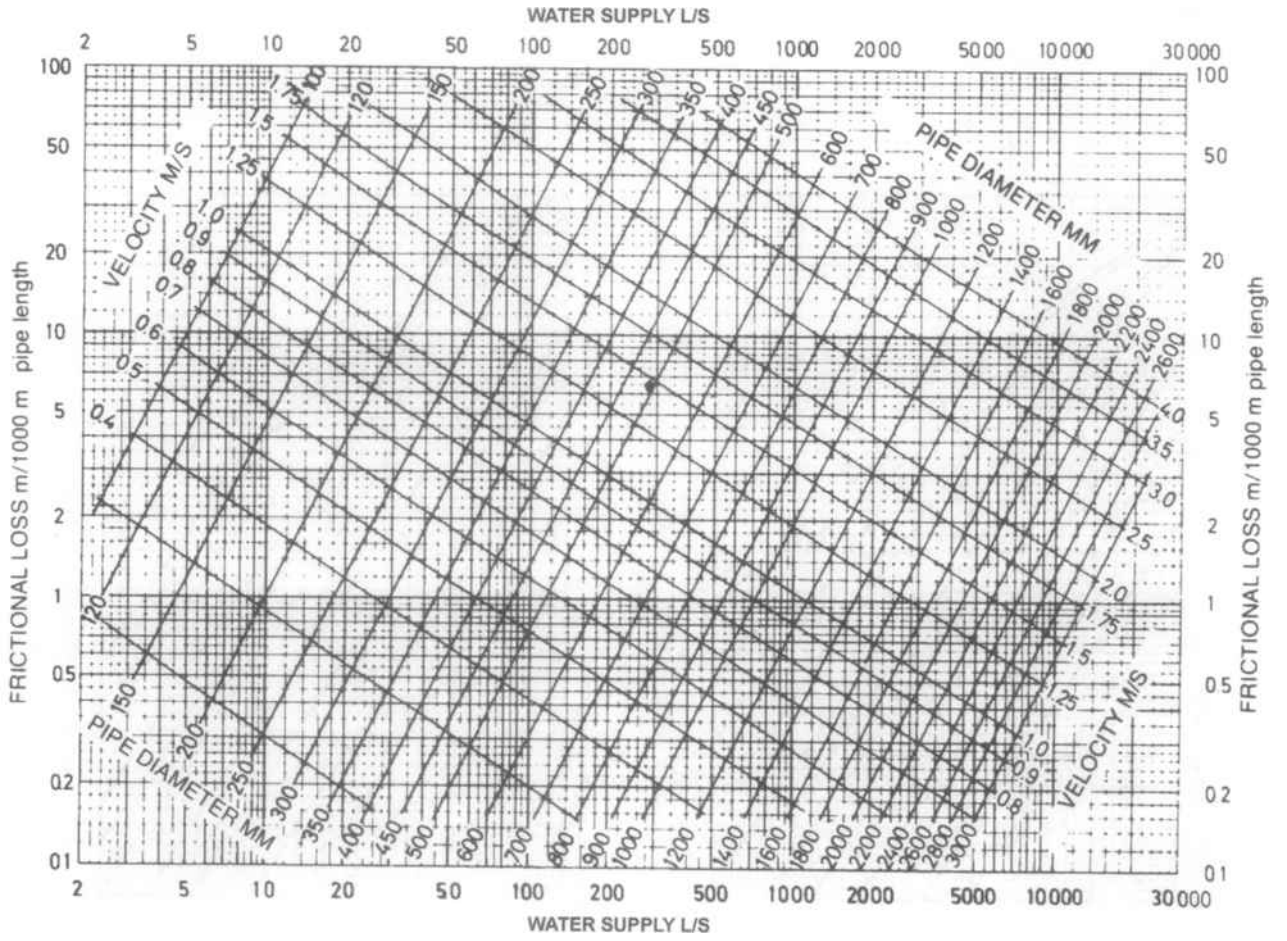

Figure 3.29 Frictional loss diagram for water at +10°C, k = 2.0 mm. Pipe type: cast iron

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3.4.3 Oil

The diagrams apply equally for other Newtonian liquids at the stated viscosity.

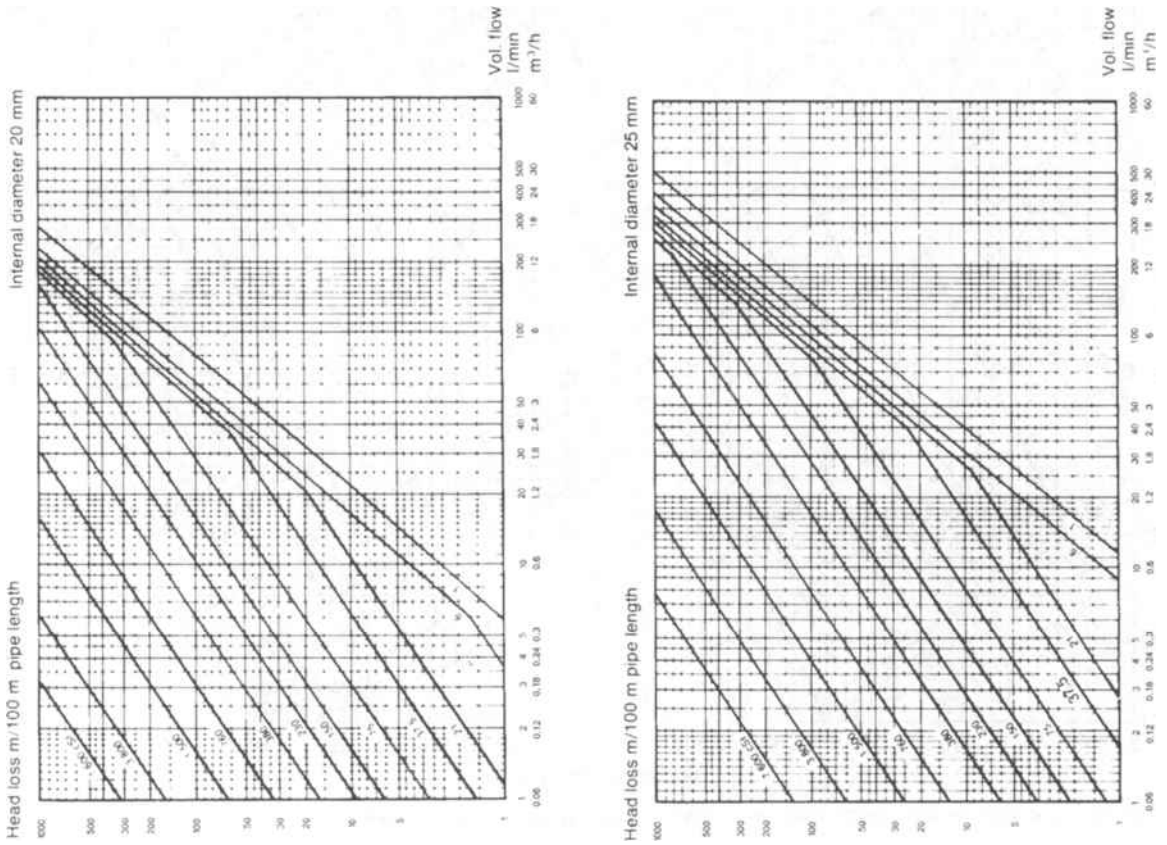

Figure 3.30 Frictional loss diagrams for oil at various viscosities, k = 0.03 mm, steel pipe

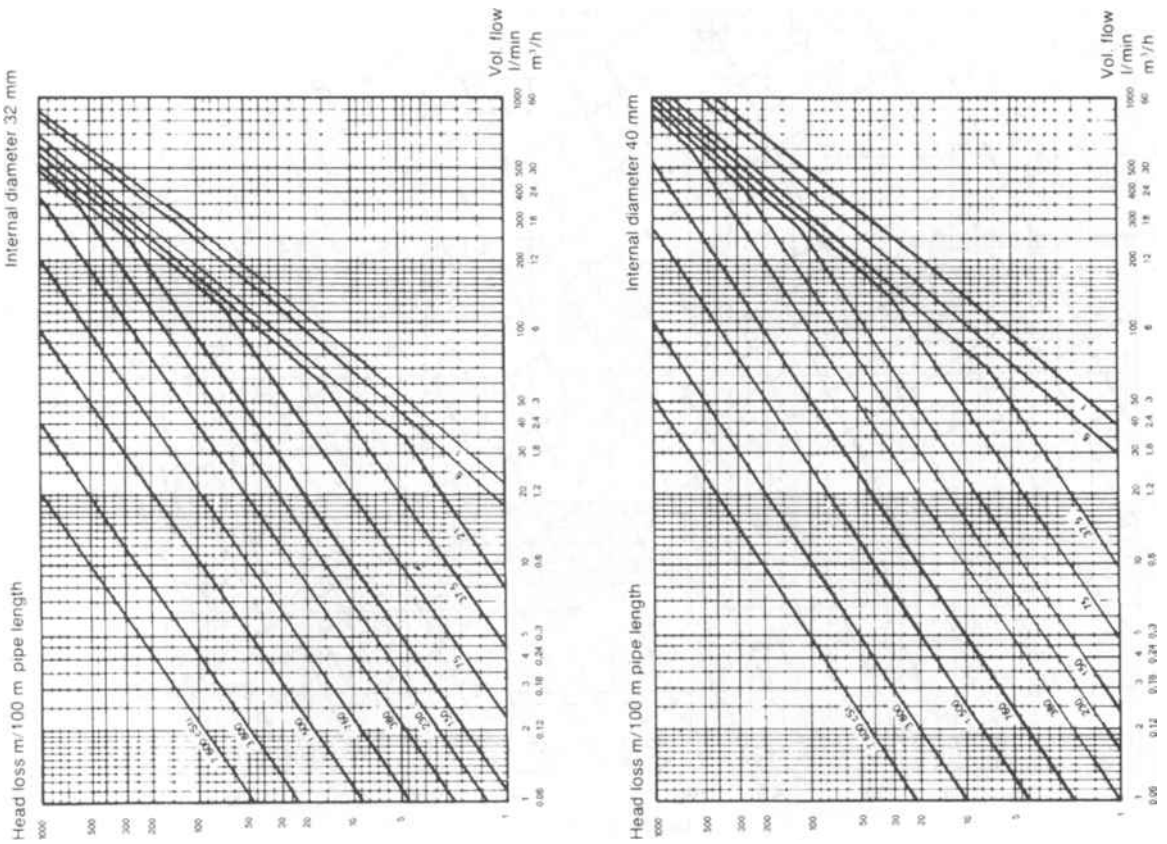

Figure 3.31 Frictional loss diagrams for oil at various viscosities, k = 0.03 mm, steel pipe

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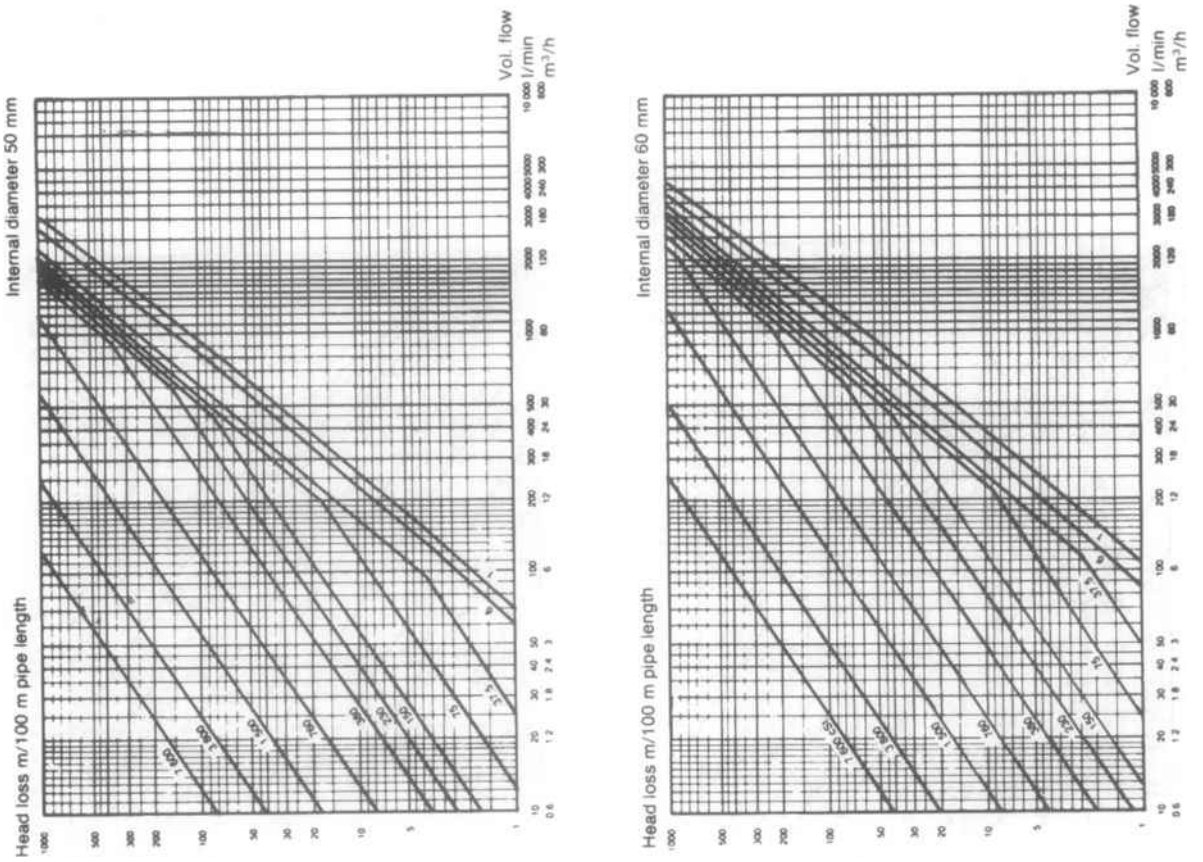

Figure 3.32 Frictional loss diagrams for oil at various viscosities, k = 0.03 mm steel pipe

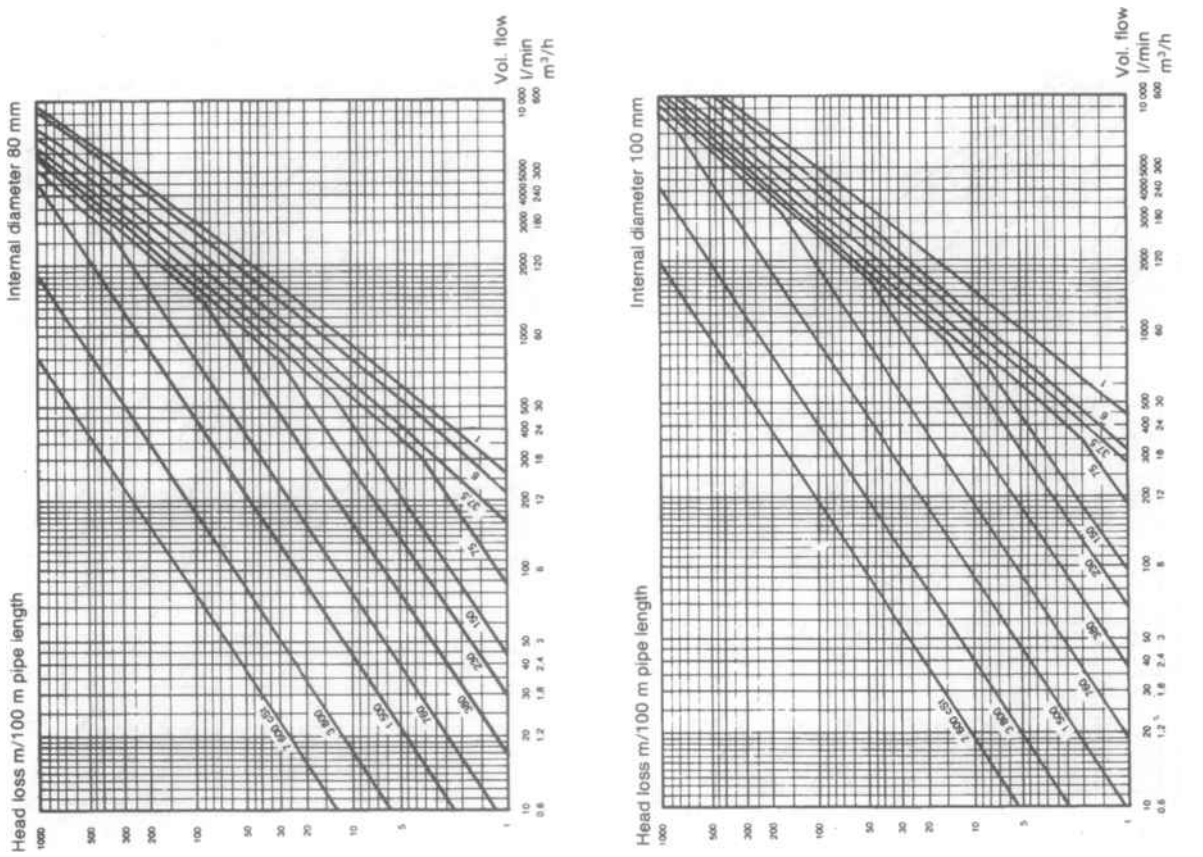

Figure 3.33 Frictional loss diagrams for oil at various viscosities, k = 0.03 mm, steel pipe

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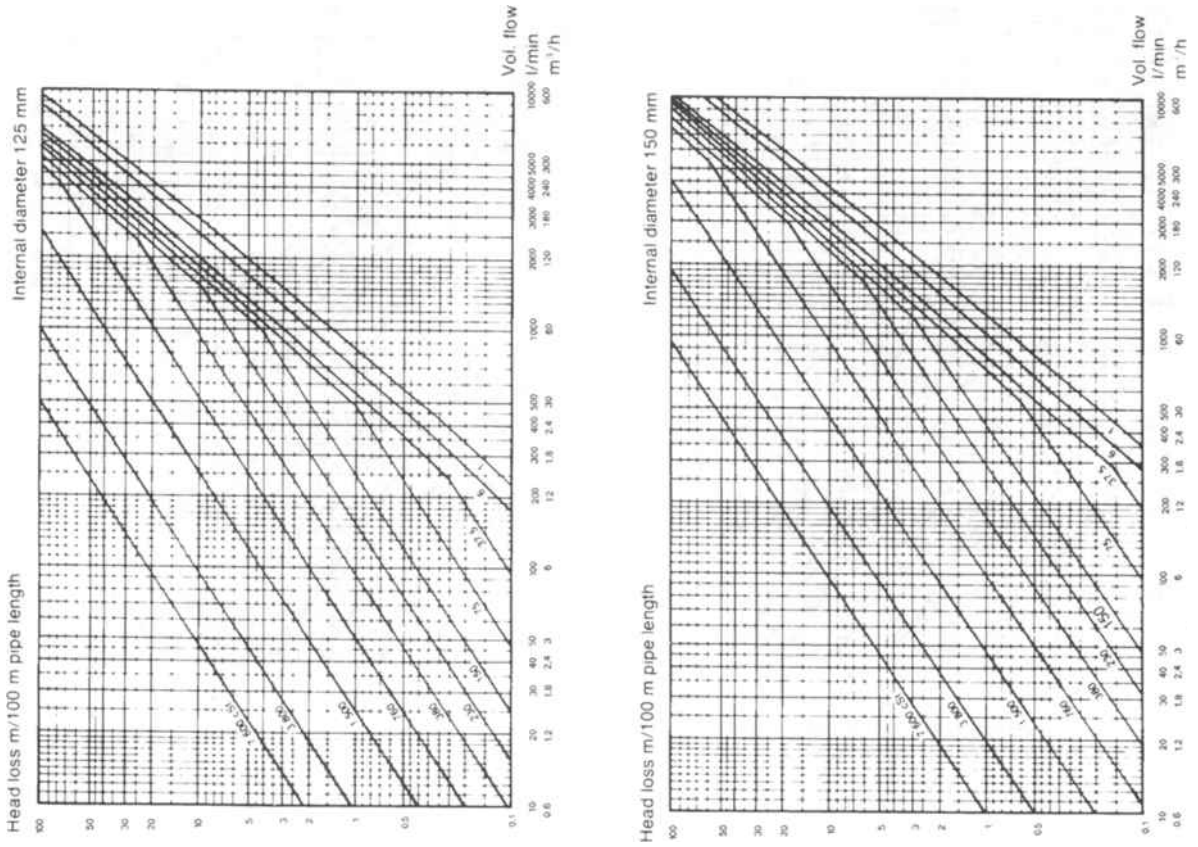

Figure 3.34 Frictional loss diagrams for oil at various viscosities, k = 0.03 mm, steel pipe

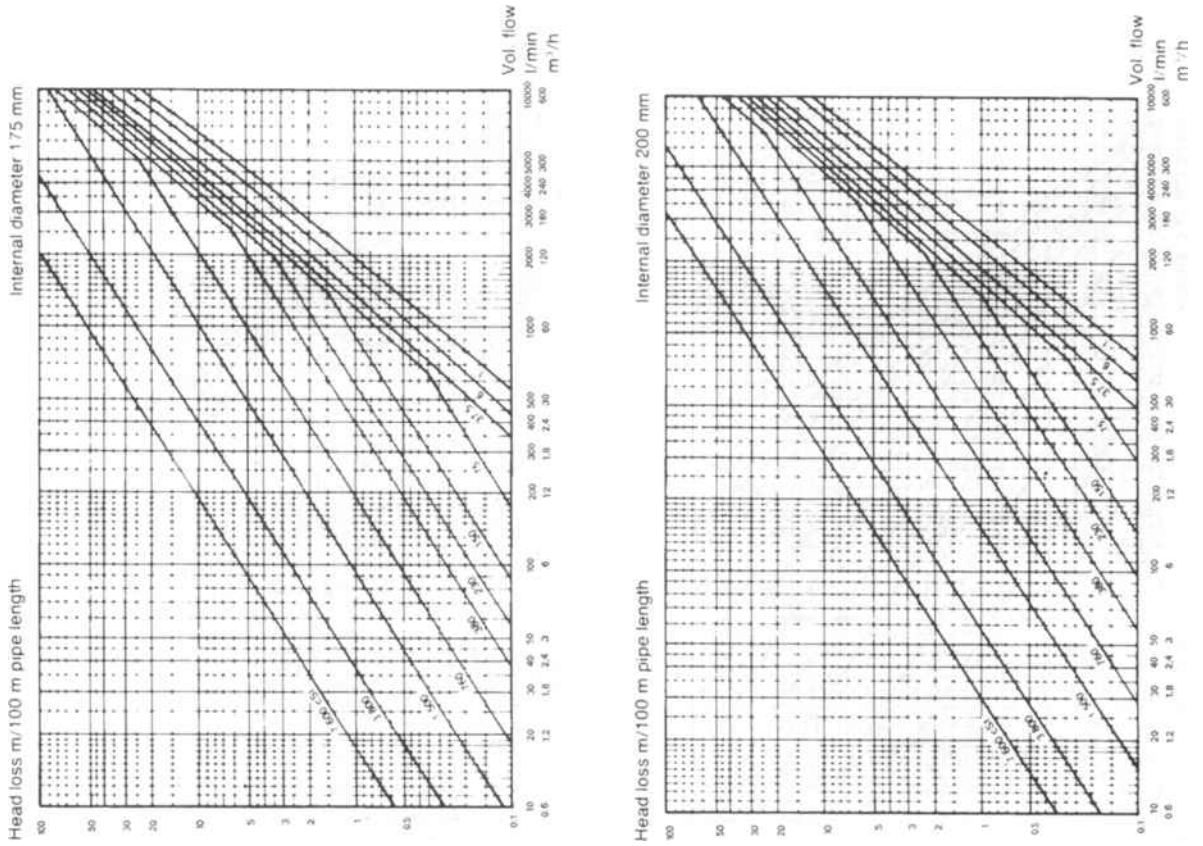

Figure 3.35 Frictional loss diagrams for oil at various viscosities, k = 0.03 mm, steel pipe

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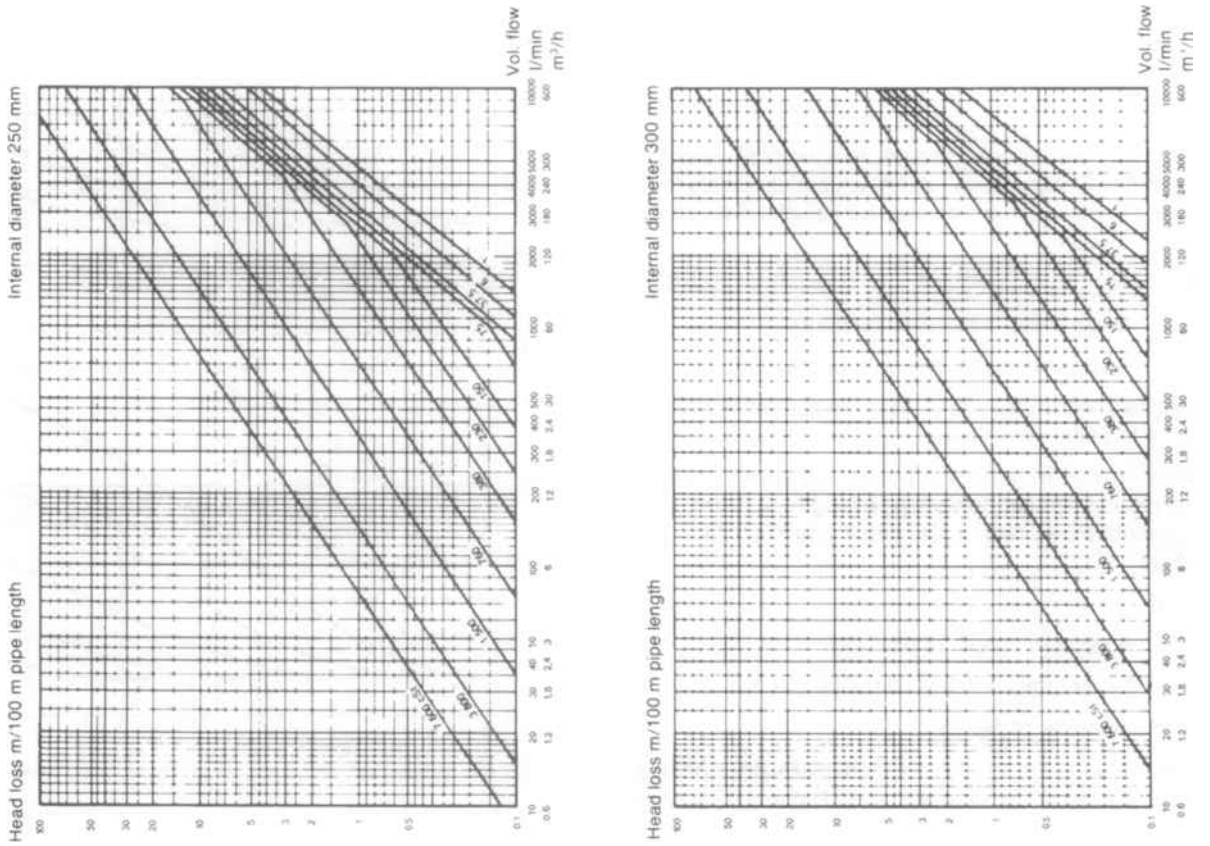

Figure 3.36 Frictional loss diagrams for oil at various viscosities, k = 0.03 mm, steel pipe

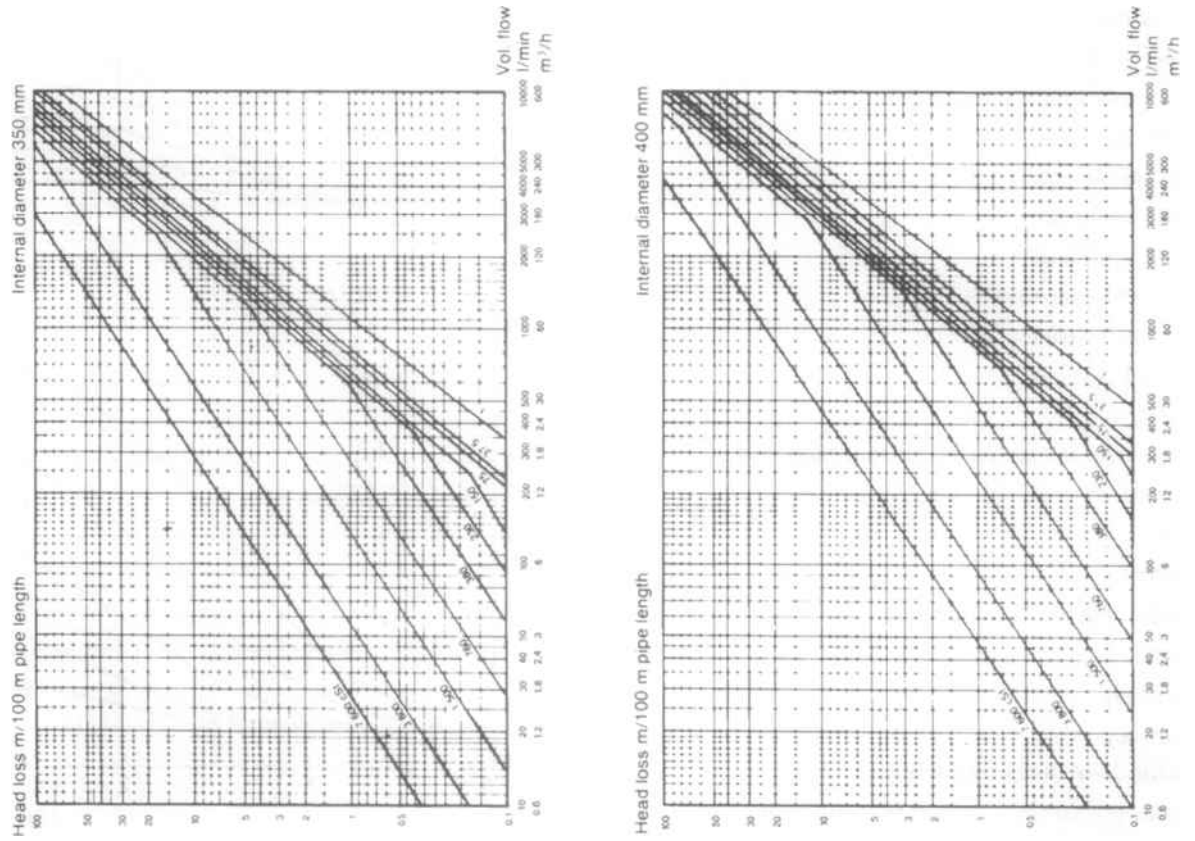

Figure 3.37 Frictional loss diagrams for oil at various viscosities, k = 0.03mm, steel pipe

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3.4.4 Paper pulp

The diagrams in Figures 3.38 and 3.39 apply for paper pulp of normal quality in stainless steel pipe:

Surface roughness of pipe 0.045 mm
Undried pulp:
Fibre ratio 75
Fineness grade 750 C.s.f.
Temperature 40°C

The diagrammatic values are computed in accordance with the method set out in Section 3.3.4. If the actual pulp differs from normal pulp, the frictional losses may be assessed using the formula:

$$h_f = f_F \cdot f_M \cdot f_D \cdot h_{fN} \qquad \text{Equ 3.38}$$

h_{fN} = loss of head for normal pulp (m)

f = factors in accordance with Section 3.3.4.

It is not possible, on the other hand, to make corrections for the roughness of the pipe in this way.

The guideline values for losses in bends and fittings, are expressed by means of an equivalent pipe length - i.e. the extra length of straight pipe which would result in a corresponding pressure loss.

$$l_{eq} = L_e \cdot d \qquad \text{Equ 3.39}$$

l_{eq} = equivalent length of pipe (m)

L_e = factor in accordance with Figure 3.38

d = diameter of pipe (m)

Since the equivalent pipe length is related to straight pipe flow in principle, it is only valid for normal pulp and cannot be applied to other liquids in other pipes.

3.5 Flow measurement

3.5.1 General

3.5.1.1 Measurement requirements

The purpose of flow measurement may be:

- For pump performance monitoring
- For pump control
- For leakage detection
- To obtain sizing data
- To obtain costing data
- To obtain productivity data

The measuring installation may be:

- Permanent
- Temporary

The measurement requirements during testing may be:

- Continuous
- Periodic
- Intermittent

Measurement data may be needed:

- Currently
- Continuously
- Cumulatively

3.5.1.2 Methods of measurement

When choosing a suitable method of measurement, account must first be taken of the category of liquid which is to be measured:

- pure liquid compounds free of contamination,

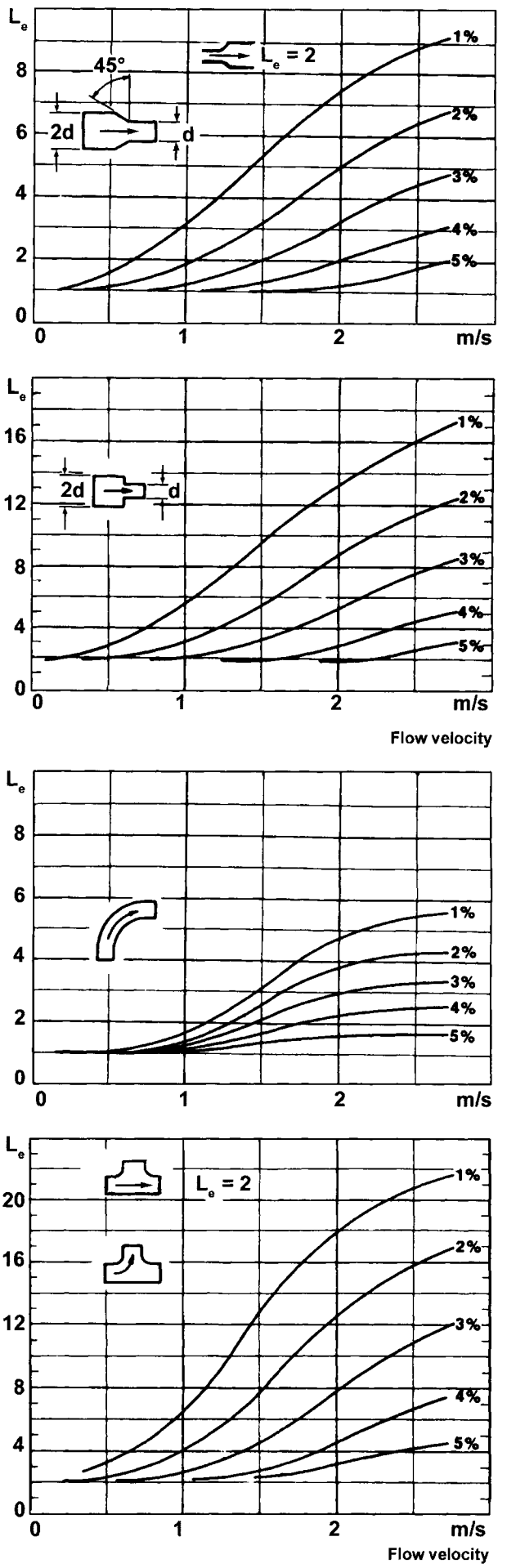

Figure 3.38 Factor for calculating equivalent pipe lengths for normal pulp in stainless steel pipe

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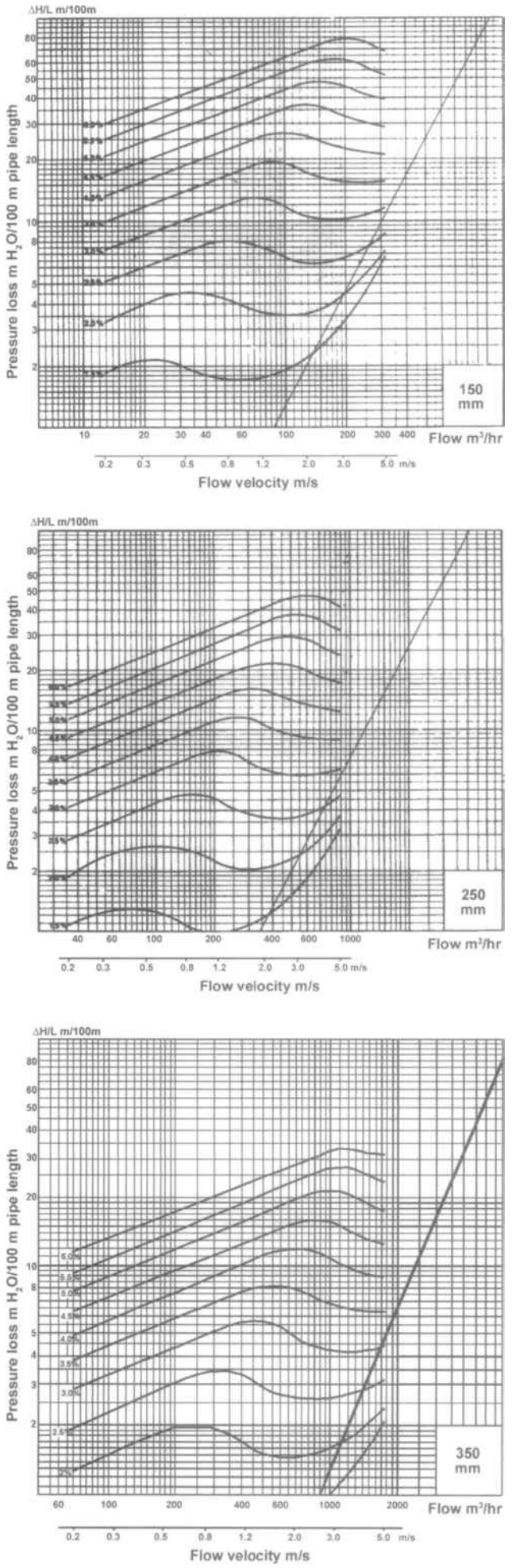

Figure 3.39 Frictional loss diagrams, normal pulp in stainless steel pipe

- contaminated liquids with a content of dissolved or undissolved; liquids, solids or gases,
- liquids having special characteristics, such as high viscosity, hot, reactive, corrosive, explosive or toxic.

Pure liquid compounds with no solid particle content or gas bubbles can be measured using most measurement equipment commercially available. The exceptions are extremely pure liquids with measuring processes which make use of contaminants normally occurring in liquids. For example, distilled water flow cannot be measured with laser or electromagnetic flow meters.

When measuring liquids contaminated with solid particles, equipment with narrow passages and sludge trapping cavities are unsuitable. When measuring liquids with special characteristics the equipment must be selected to obtain compatible performance and material combinations having due regard to the particular properties of the liquid. Laser, electromagnetic, ultrasonic and turbine devices are more suitable than others, for hazardous liquids, because of the absence of external pipe connections.

3.5.1.3 Measuring range

Instruments should be selected so that the main area of operation falls within 20% to 60% of their full-scale range.

Too large a scale may mean that a large part of the measurement occurs below 20% capacity, with resultant errors in measurement.

Too small a scale may cause the capacity of the meter to be exceeded, which will also give incorrect readings. In order to avoid measuring in unsuitable ranges, the measuring equipment in some cases may be fitted with several scales in order to minimize measurement errors.

3.5.1.4 Accuracy of measurement

Accuracy is usually stated in percentage error of full scale and applies in ideal conditions as regards installation of the equipment according to the manufacturers instructions.

It can be seen from Figure 3.40 that in the region below 10% of maximum value, the measurement accuracy is obviously poor. Within the normal measurement range, the errors extend from ± 8 to 25% of the actual measurement value, despite the claim that the meter has a measurement error within ± 5% at full-scale deflection. The absolute error is assumed to be ± 5% of maximum value within the complete measuring range.

3.5.1.5 Straight pipe length

Most flow meters require a well developed velocity profile ahead of the measuring instrument. Requirements are often placed on the minimum length of straight pipe in order to maintain the guaranteed accuracy. The necessary straight pipe lengths, in many cases, may be difficult or expensive to arrange. However, pump manufacturers will not be pleased if the

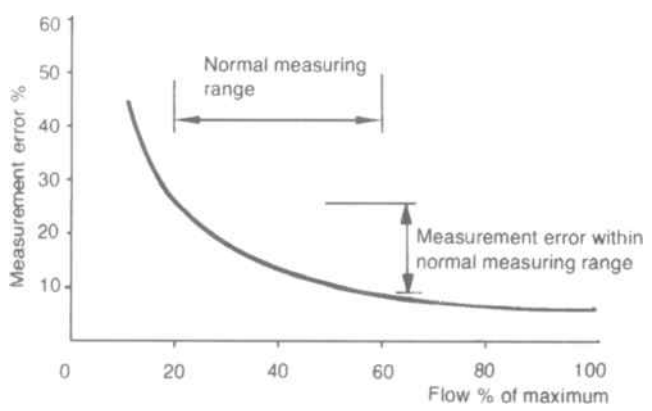

Figure 3.40 Relative measurement error with actual flow

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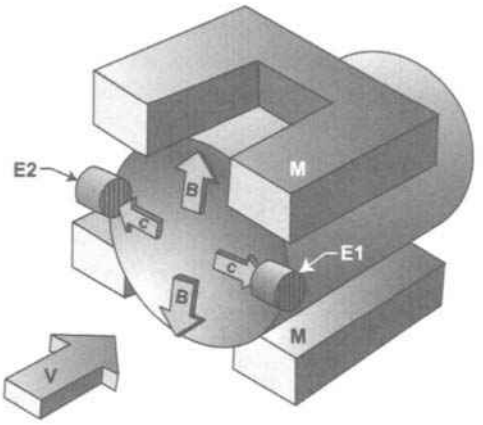

Figure 3.41 Electromagnetic flow meter principle
Courtesy of Liquid Controls

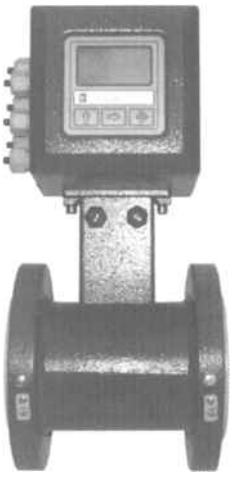

Figure 3.43 Typical electromagnetic flow meter
Courtesy of Liquid Controls

quality of their products are impugned by poorly installed flow meters. Different measuring instruments give rise to considerably varying flow losses in the installation. Loss of head associated with flow measurement can be responsible for undesirable additions to the pump energy requirement and power consumption.

3.5.2 Flow meters for pipes

This Section outlines the function and use of a number of varied flow meters. The typical properties and characteristics of these in pipes are summarised in Table 3.7; the characteristics quoted, varying of course in the case of special types.

3.5.2.1 Electromagnetic flow meters

Electromagnetic flow meters consist of a sensor and an integrally mounted or remote mounted converter. Figure 3.41 illustrates the principle and Figure 3.42 shows a typical exterior view. It consists of a lined flow tube, two electromagnetic coils (M), and two diametrically opposed electrodes (E1 & E2). Faraday's Law of Electromagnetic Induction is the basis for magnetic flow measurement. The law states that when a magnetic field is created at right angles to the flow tube through which a conductive liquid is flowing, the voltage induced in the electrodes (immersed in the fluid) is proportional to the velocity of the liquid. This induced voltage is converted to a scaled digital pulse output and, optionally, to an analog output. The analogue output signal is a linear output directly proportional to rate of flow.

As the output signal is directly proportional to the flow there is no need for linearisation. This means that it can be switched simply between different display scales. It can even be arranged so that it switches automatically to a more sensitive scale as soon as the flow drops below a certain limit, for example, 20% of full scale. The electromagnetic flow meter does not recognize the direction of flow through the meter head. This can cause problems if, for example, the meter is fitted to a pump line from a pump station which uses reverse flushing with product to clean out the pump. The reverse flush quantity will then have been measured three times.

The process liquid must be conductive to some extent; liquids as low as 0.008 μS/cm have been measured successfully. Because the device is electrical, certification is advisable for hazardous area.

Method	Liquid	Measurement range	Error	Straight pipe requirements	Head loss	Comments
Electromagnetic	(1)	≤ 40:1	± 0.25% possible	5D	very low	liquid must have electrical conductivity
Positive displacement	(2)	≤ 100:1	± 0.15% possible	short	moderate	various types; vane, love, rotary piston, screw
Orifice plate	(3)	4:1	2% of reading	25D	large	easily manufactured those to ISO R541 need no calibration
Nozzle	(3)	6:1	2% of reading	25D	large	those to ISO R781 need no calibration
Venturi meter	(3)	8:1	2% of reading	25D	small	
Multiple Pitot tube	(4)	10:1	± 0.5% after insitu calibration ± 1.1% uncalibrated	8D to 30D	small	
Axial turbine	(4)	20:1	0.25% FSD or 1% of reading	25D	small	
Pelton Wheel	(4)	≤ 200:1	± 0.5%	short	small	0.01 l/m to 1200 m^3/h
Ultrasonic Doppler	(5)	50:1	±2%	20D	very small	
Ultrasonic timing	clean	10:1	±0.5%	20D	very small	
Swirl	clean	10:1	1.5% FSD or 2.5% of reading	15D	moderate	
Vortex shedding	(6)	15:1	±0.65% digital ±1.35% analogue	15D	moderate	
Variable orifice	(7)	10:1	1.5% FSD	short	moderate	
Calorimetric	clean	100:1	± 5%	short	moderate	low flows

FSD = Full scale deflection

(1) Liquid can contain solid particles and gas bubbles

(2) Clean liquid is best, solids cause wear, some viscosity desirable for lubrication and to reduce slip

(3) Solids damage orifice and reduce accuracy, clean liquid preferred, range limited by Reynolds Number

(4) Clean liquid, range limited by Reynolds Number

(5) Liquid must contain impurities

(6) Slightly contaminated possible, limited by Reynolds Number

(7) Clean liquids, high viscosity possible

Table 3.7 Typical properties of flow meters in pipes

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Figure 3.43 Typical positive displacement flow meter
Courtesy of Liquid controls

3.5.2.2 Positive displacement (PD) flow meters

Most positive displacement pumps, when driven as a "motor", i.e. with reversed direction of flow, work like turbines. There are as many positive types of mechanical flow meter, as there are positive displacement pumps, adapted to various liquid characteristics and capacities. A typical example is shown in Figure 3.43. This PD meter is used for refined fuels applications.

Some types, such as rotary piston flow meters, rely on the process liquid viscosity for lubrication and sealing, minimum viscosity of the order of 20 cP. Other positive displacement flow meters can cope with low viscosity non- lubricating liquids, like petrol, and very viscous liquids, over 300000 cP. Flow rates as low as 0.002 l/m can be measured and working pressures over 500 bar can be accommodated. Larger meters can measure flows of 3700 l/m at pressures up to 100 bar. PD flow meters can be completely mechanical, with no external power source required, and are eminently suitable for remote or hazardous installations.

3.5.2.3 Orifice plates and nozzles

Measuring flanges and orifices are dimensioned in accordance with ISO R541. In addition, the ISO Standard provides information about installation requirements and the calculation process for conversion of the pressure differential into flow. The guaranteed degree of accuracy is also specified. See Figure 3.44.

The range of measurement is limited at the lower end by increasing measurement errors at low Reynolds Numbers and at the top end by sharply rising head losses. Orifice plates and nozzles are simple devices with no moving parts. Problems can arise with the small bore instrument pipework connecting the manometer.

Normal straight pipe lengths are of the order of 20 x pipe diameter upstream of the orifice or nozzle and 10 x pipe diameter downstream.

As can be seen from Figure 3.45, the nozzle has lower pressure losses than the orifice plate. From this point of view, the best differential pressure flow meter is the venturi meter.

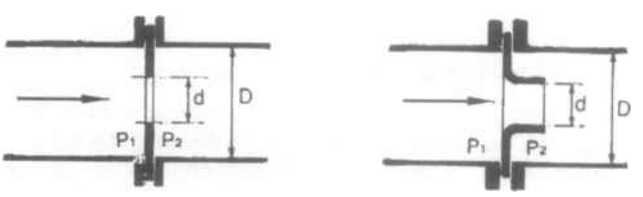

Figure 3.44 Orifice plate and nozzle. The flow is proportional to the square root of the pressure differential

Diameter ratio d/D	Loss coefficient ζ		
	Orifice	Nozzle	Venturi
0.4	85	30	6
0.5	30	10	2
0.6	12	3	1
0.7	5	1	0.5
0.8	2	0.5	0.2

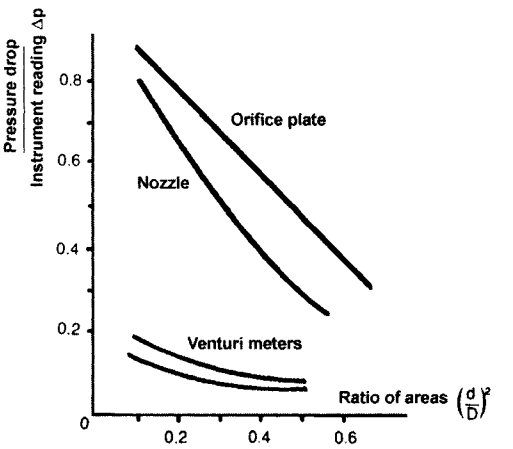

Figure 3.45 The pressure drop for a differential pressure flow meter expressed as a proportion of the instrument reading and as a loss coefficient.

3.5.2.4 Venturi meters

Venturi meters are available in long and short versions, see Figure 3.46. The lowest flow losses occur with the long version. Venturi meters are covered in ISO 5167-1 as regards dimensions, installation, calculation procedure and degree of accuracy.

3.5.2.5 Turbine meters

A turbine meter consists essentially of a turbine impeller driven by the flow of liquid, see Figure 3.47. Either the number of revolutions is measured, as for summation measurements, or the rpm (pulses per unit time) in volume flow measurements. Turbine meters are available in both axial flow and Pelton Wheel versions. Axial flow meters can be very large whereas Pelton Wheel meters are particularly suitable for very small flows with high accuracy.

Axial flow turbine meters can have aerofoil profile blades and specially shaped stators to obtain very high degrees of accuracy and are often used for calibration purposes. Simpler axial turbine meters, the so-called Woltmann meter, are not shaped for highest accuracy, however, but for general sensitivity.

3.5.2.6 Ultrasonic meters

Flow measurement using ultrasonic sound waves can be performed in two different ways and require different liquid charac-

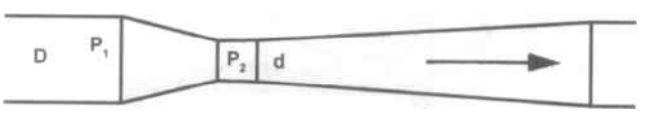

Figure 3.46 Venturi meter, long version

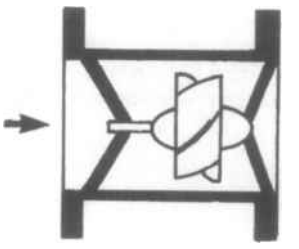

Figure 3.47 Simple turbine meter

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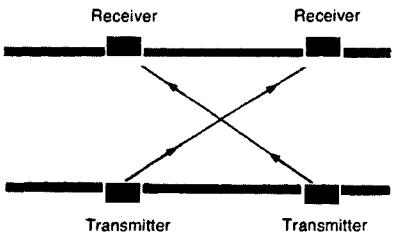

Figure 3.48 Flow measurement, ultrasonic

teristics to work correctly.

The first method requires the liquid to have some impurities, like solids or gas bubbles, to reflect the sound waves. The injected ultrasonic signal is reflected and scattered by impurities. The receiver detects the returning signals and uses Doppler techniques to convert the frequency shift into liquid velocity. The frequency shift is directly proportional to flow.

Pure clean liquids can be measured by a different method. Two transmitters inject sound waves, one in the direction of flow, and the other against the direction of flow. The difference in transit time between transmitter and receiver for these two signals is a measure of the velocity of flow and therefore of the flow itself. The difference in transit time can be converted to frequency, this being directly proportional to the flow. The frequency difference is independent of the speed of sound in the liquid. The principle is illustrated diagrammatically in Figure 3.48.

3.5.2.7 Swirl flow meters

A set of fixed vanes impart a tangential component of velocity, rotational component, at the same time as creating a secondary flow. The rotating secondary flow gives rise to pressure pulsations which are detected by a fixed sensor in the smallest section. The number of pulsations per unit time is a measure of the flow. This meter, which is shown in Figure 3.49, contains no moving parts.

3.5.2.8 Vortex shedding flow meters

When liquids flow round specially shaped bodies, there is a periodic release of vortices trailing behind the bodies. Since the boundary layer release changes from one side of the body to the other, the pressure distribution also changes around the body. The number of changes per unit time is proportional to the flow. This type of flow meter can contain no moving parts depending upon how the vortices are counted. It is shown diagrammatically in Figure 3.50.

The vortices can be counted by a thermistor detecting the cooling or heating effect changes caused by the different flow patterns. As the vortices are shed alternately from each side the

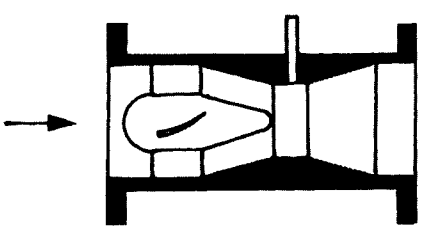

Figure 3.49 Swirl flow meter

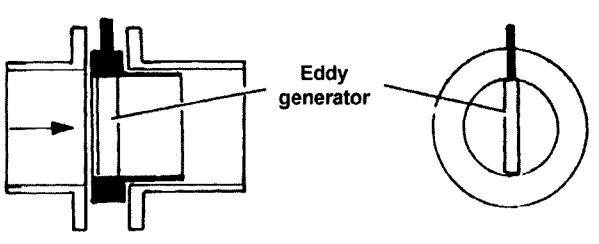

Figure 3.50 Vortex shedding flow meter

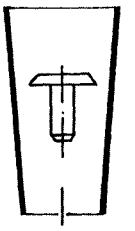

Figure 3.51 Variable orifice flow meter

pressure differential across the body changes direction. The direction of the pressure gradient can be detected by a moving sphere or disc. The vortices can be counted by a thin vane, downstream of the body, deflecting from side to side. Lastly, the vortices can be counted by ultrasonic sound waves.

3.5.2.9 Variable orifice or aperture flow meters

With this type, a spring loaded restriction, shaped like a poppet valve and mounted in a fixed orifice, moves to adjust the flow area in response to the liquid velocity. The flow can be read directly from the position of the restriction. Variable aperture flow meters are available in various sizes, and flows over 1400 m^3/h can be accommodated. See Figure 3.51.

3.5.2.10 Calorimetric flow meters

Calorimetric flow meters work on the principle of applying a known quantity of heat to the flowing liquid and measuring the consequential temperature rise. This technique is most useful in small flow rates, up to 10 l/min.

3.5.3 Flow meters for open channels

Useful data for the various methods described in this Section is given in Table 3.9.

Method of measurement	Liquid	Range of measurement	Degree of accuracy	Remarks
Flume	Clean to heavily contaminated	10:1	5% of max.	Several shapes available. Low head loss
Weir Notch	Clean to slightly contaminated	10:1	2% of max.	Several shapes available. Moderate loss of head. Risk of sludging
Vane	Clean	High ~ 20:1	0.5% of max.	

Table 3.9 Typical data for open channel flow meters

3.5.3.1 Measurement flumes

Measurement flumes are based on the idea that when the flow of water is so constrained a specific relationship will exist between flow and water level upstream of the flume, or, in some designs, between flow and difference in water level along the length of the flume. Flow measurement is thus converted to a level measurement.

The most common forms of measurement flume are the Parshall flume (see Figure 3.52) and the Palmer-Bowlus flume. The relationship, water level to flow, is determined empirically and therefore each flume has to be an exact copy of the master flume from which the calibration curve is derived. The degree of accuracy is stated in various sources as being in the order of ±3% to ±5% for precision made flumes and measurement devices.

3.5.3.2 Weirs and notches

These are illustrated diagrammatically in Figure 3.53. Weirs consist of a wall, of various proportions, covering the full width of the channel. Theoretical expressions can be derived, from Bernoulli's equation, for the relationship between the head above the weir to the flow rate. Theoretical expressions have proved very inaccurate and calibration of the finished installa-

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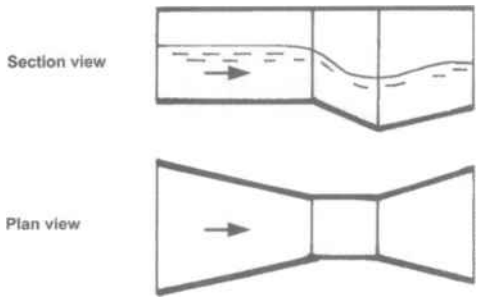

Figure 3.52 Parshall flume

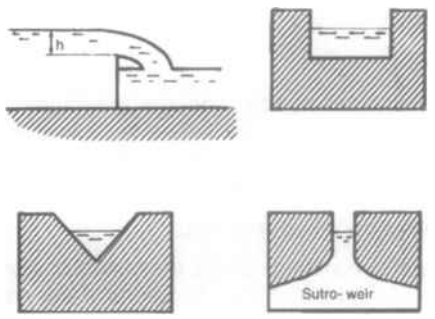

Figure 3.53 Different types of weir and notch for flow measurement

tion is essential. Empirical equations are available for designing weirs but calibration is still imperative after construction for accurate measurements. These are known as:

Francis's Formula

$$Q = 1.84(L - 0.2H)H^{1.5} \qquad \text{Equ 3.40}$$

Bazin's Formula

$$Q = \left(0.4046 + \frac{0.003607}{H}\right)\sqrt{2g}LH^{1.5} \qquad \text{Equ 3.41}$$

where:

L	=	length of weir (m)
H	=	height of liquid (m)
g	=	gravitational acceleration (m/s^2)

Notches consist of an accurately shaped sharp-edged "partial orifice" over which the liquid flows. A notch can be rectangular or triangular in shape. The flow rate is directly related to the height (H) of the upper liquid surface above the notch. The coefficient of discharge, C_d, must be evaluated for the specific installation.

Rectangular notch

$$Q = \frac{2}{3}C_d\sqrt{2g}LH^{1.5} \qquad \text{Equ 3.42}$$

Vee notch

$$Q = \frac{8}{15}C_d\sqrt{2g}\tan\frac{q}{2}H^{2.5} \qquad \text{Equ 3.43}$$

where:

L	=	length of weir (m)
H	=	height of liquid (m)
g	=	gravitational acceleration (m/s^2)
C_d	=	coefficient of discharge
θ	=	angle of vee notch (degree)

The Sutro-weir is not an overfall weir in the strict sense of the word. It is, however, a simple arrangement for flow measurement of heavily contaminated liquids. The Sutro-weir does not have sludge pockets and gives a linear relationship between upstream liquid level and flow.

3.5.3.3 Vanes

A very accurate flow measurement can be carried out using a measurement vane. Its use is shown diagrammatically in Figure 3.54.

The vane, which is shaped to fit the channel, moves with the liquid flow. The time is measured to travel between two predetermined positions.

3.5.4 Other instruments for flow measurements

3.5.4.1 Tanks

Measuring the time taken to fill or empty a tank of known volume is a well tried method of flow measurement. As an alternative to volume measurement, the tank can be weighed before and after filling. Properly performed, the weighing method can offer the most accurate of all known methods and is therefore often used for calibration. Both volume and weighing measurement methods are ideal when the flow is not constant, from reciprocating pumps for example.

When the weighing method is used and the results subsequently converted to volume, care should be exercised during the conversion. Ensure all parties concerned are using the same, agreed, liquid data. Even for water.

3.5.4.2 Pitot-static tube

Velocity of flow can be determined by the use of the Pitot-static tube (see Figure 3.55). Because of the velocity profile in pipe-lines and ducts, the velocity at a particular point cannot be converted to a flow. By measuring the dynamic pressure at a number of points the flow can be summated with greater accuracy. Various means of measurement with several outlet pressure points distributed around the pipe or duct section are available commercially.

3.5.4.3 Current meter

A current meter is a device used in large channels, canals and rivers, to measure the current speed at a particular position. Some instruments are mounted on traversing gantries so that a velocity distribution across the section can be averaged. The device converts the linear motion of the liquid into rotary motion which can be measured. Two designs are available. Figure 3.56 shows a propeller version, the rotary speed of which varies linearly with the velocity of flow. The other version consists of a simple Pelton Wheel device which uses hollow cones as buckets and is mounted with the axis of rotation vertical.

Figure 3.54 Flow measurement using vanes

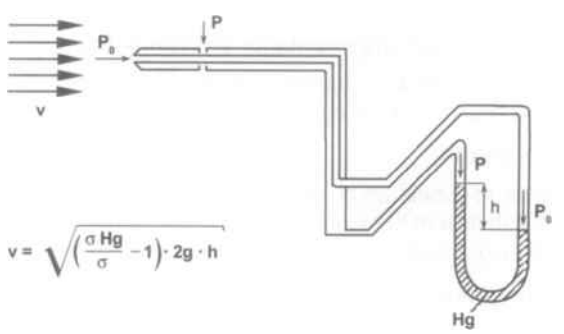

Figure 3.55 Pitot-static tube with U-tube manometer

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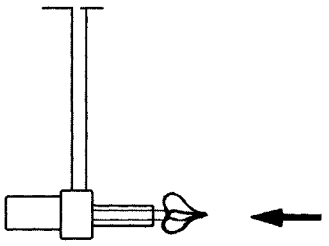

Figure 3.56 Propeller current meter

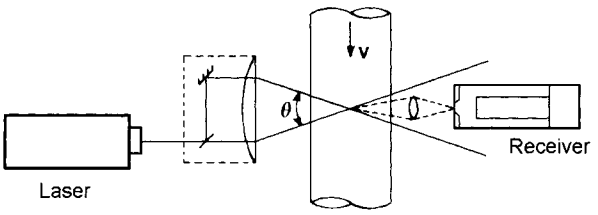

Figure 3.57 Flow measurement using lasers

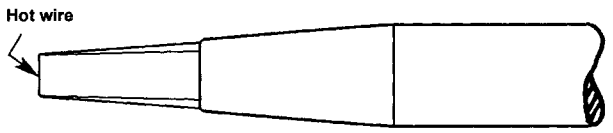

Figure 3.58 Simple probe for hot-wire anemometer

3.5.4.4 Laser-doppler

A laser beam is divided into two beams in a prism as outlined in Figure 3.57. The beams are focussed together again in the volume to be measured in the flow field, where an interference pattern is generated. A particle crossing the interference band reflects light pulses to a receiver. The frequency of the reflected light is a measure of the velocity of flow. The measuring equipment is very accurate but costly and its use is limited to the laboratory. See Figure 3.58.

3.5.4.5 Hot-wire anemometer

The hot-wire anemometer is the most well known thermal anemometer, and measures a liquid velocity by noting the heat convected away by the liquid. See Figure 3.58. The core of the anemometer is an exposed hot wire either heated up by a constant current or maintained at a constant temperature. In either case, the heat lost to fluid convection is a function of the fluid velocity. (As an alternative to wire, thin film may also be used. The meter is delicate and limited to laboratory use.)

By measuring the change in wire temperature under constant current or the current required to maintain a constant wire temperature, the heat lost can be obtained. The heat lost can then be converted into a fluid velocity in accordance with convective theory.

3.5.4.6 Various arrangements for checking flow

There are a number of methods, which are intended rather for checking flow than for accurate flow measurements:

- **Pipe bend**

  Pressure outlets are placed perpendicular to the direction of through flow at the inner and outer bend radii respectively. The error can be limited to ±10 % for a calibrated bend.

- **Sight glass**

  The position of a pivoted body in the flow (flap, paddle, etc.) can be observed through a viewing glass.

- **Pump curve**

  The flow can be assessed from pressure measurements and via the known H-Q curve of the pump. This method must be used with caution. A knowledge of how the pump performance has changed, since new, is very important.

- **Free jet**

  The outlet velocity of a free jet can be estimated by measuring the length of throw, and with the velocity the flow.

- **Floats**

  The time taken for a float, or other trace substance, to travel a given length gives a good idea of the flow in an open channel of known section.

- **Tracers**

  Trace substances can be used in pipe systems. An example is salt dosing in pulses. The salt cloud is traced by means of electrodes which detect the changed electrical conductive properties of the water.

- **Concentration measurements**

  With a known constant dosing, the concentration in the liquid is dependent on the flow. Solutions of salt in water have been used with good results.

- **Pump speed**

  For speed regulated pumps, the measurement of pump speed provides a means of checking the flow. If both the pump and system curve are known, this method can provide reasonable accuracy. Again wear in the pump and it's effect on pump performance must be considered. Positive displacement pumps produce a relatively constant output over a wide pressure range. System response is not so important in these situations.

3.5.5 Measurement standards

Methods of measurement of liquid flow are dealt with in various standards.

3.5.5.1 Measurement in pipe and duct systems

BS 1042-2.1:1983, ISO 3966:1977 Measurement of fluid flow in closed conduits. Velocity area methods. Method using Pitot static tubes.

BS 1042-2.2:1983, ISO 7145:1982 Measurement of fluid flow in closed conduits. Velocity area methods. Method of measurement of velocity at one point of a conduit of circular cross section.

BS 1042-2.3:1984, ISO 7194:1983 Measurement of fluid flow in closed conduits. Velocity area methods. Methods of flow measurement in swirling or asymmetric flow conditions in circular ducts by means of current-meters or Pitot static tubes.

BS 1042-1.4:1992 Measurement of fluid flow in closed conduits. Pressure differential devices. Guide to the use of devices specified in Sections 1.1 and 1.2. 1.

ISO TR 3313:1998 Measurement of fluid flow in closed conduits — Guidelines on the effects of flow pulsations on flow-measurement instruments.

BS 5857-1.5:1980, ISO 2975-VII:1977 Methods for measurement of fluid flow in closed conduits, using tracers. Measurement of water flow. Transit time method using radioactive tracers.

BS 5857-2.1:1980, ISO 4053-I:1977 Methods for measurement of fluid flow in closed conduits, using tracers. Measurement of gas flow. General.

BS 5857-2.4:1980, ISO 4053-IV:1978 Methods for measurement of fluid flow in closed conduits, using tracers. Measurement of gas flow. Transit time method using radioactive tracers.

BS 5857-1.4:1980, ISO 2975-VI:1977 Methods for measure-

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ment of fluid flow in closed conduits, using tracers. Measurement of water flow. Transit time method using non-radioactive tracers.

BS 5857-1.1:1980, ISO 2975-I:1974 Methods for measurement of fluid flow in closed conduits, using tracers. Measurement of water flow. General.

BS 5857-1.3:1980, ISO 2975-III:1977 Methods for measurement of fluid flow in closed conduits, using tracers. Measurement of water flow. Constant rate injection method using radioactive tracer.s

BS 6199-1.2:1991, ISO 9368-1:1990 Measurement of liquid flow in closed conduits using weighing and volumetric methods. Weighing method. Procedures for checking static weighing system.s

EN ISO 8316:1997 Method of liquid flow in closed conduits. Method by collection of the liquid in a volumetric tank. (See BS 5844, BS 5875, ISO 4185, ISO 4373).

EN 7405:1991 Guide to selection and application of flowmeters for the measurement of fluid flow in closed conduits.

EN 24006:1993, ISO 4006:1991 Measurement of fluid flow in closed conduits. Vocabulary and symbols.

EN ISO 5167-1:1997 Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full. Orifice plates, nozzles, and venturi tubes inserted in circular cross-section conduits running full.

EN ISO 5167-1:2003 Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full. General principles and requirements.

EN ISO 6817:1997 Measurement of conductive liquid flow in closed conduits. Method using electromagnetic flowmeters.

ISO TR 12764:1997 Measurement of fluid flow in closed conduits. Flowrate measurement by means of vortex shedding flowmeters inserted in circular cross-section conduits running full.

3.5.5.2 Measurement in open channels

BS 3680-10C:1996 Measurement of liquid flow in open channels. Sediment transport. Guide to methods of sampling of sand-bed and cohesive-bed materials.

BS 3680-3Q:2002 Measurement of liquid flow in open channels. Guidelines for safe working practice in river flow measurement.

BS 3680-8A:1989, ISO 2537:1988 Measurement of liquid flow in open channels. Measuring instruments and equipment. Current meters incorporating a rotating element.

BS 3680-7:2000 Measurement of liquid flow in open channels. Specification of equipment for the measurement of water level.

BS 3680-2A:1995, ISO 9555-1:1994 Measurement of liquid flow in open channels. Dilution methods. General.

BS 3680-4B:1986, ISO 4360-1984 Methods of measurement of liquid flow in open channels. Weirs and flumes. Triangular profile weirs.

BS 3680-3F:1986, ISO 1088-1985 Measurement of liquid flow in open channels. Stream flow measurement. Collection and processing of data for determination of errors in measurement.

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Pump theory

4

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4.1 Introduction

Use is made of pumps everywhere there is a need for liquid transportation. Sometimes it is clean liquid which is pumped, sometimes the liquid is used as a medium for carrying thermal energy or solid suspended material etc. Due to widely varying operational requirements there are a large number of different pump designs available. A classification into four main categories has been made in an attempt to provide a clear picture of the types of pumps in use today and forms the basis of the pump types described in detail in Chapter 1. The pump types in Chapter 1 should be studied individually to assist in the selection of the appropriate type.

- Rotodynamic pumps
- Special rotodynamic pumps
- Positive displacement pumps
- Other pumps

The construction of rotodynamic pumps is characterised by one or more impellers equipped with vanes which rotate in a pump casing. Those forces which come into being when liquid flows around the vanes are utilised in the conversion of energy. The rotodynamic pumps are called radial pumps or axial pumps depending on the main direction in which the liquid flows through the impellers. Alternatively, the terms "centrifugal pumps" for radial pumps, and "propeller" for axial pumps are used. Of course, intermediate forms between purely radial and purely axial pumps are also found and as such, are termed "mixed-flow" pumps. Figure 4.1 shows the most usual and most important of the rotodynamic pumps.

Certain aspects of the special rotodynamic pumps are closely related to the radial rotodynamic described above. These pumps are not true rotodynamic pumps because their performance does not meet the expectations predicted by the rotodynamic equations. The kinetic energy of the liquid is transformed into static head but the actual pump performance must be confirmed by testing. The peripheral pump uses impellers with vanes to increase the liquid kinetic energy whereas the Pitot tube pump utilises a rotating drum. See Figure 4.2. Both pumps are very useful because the differential head increase is more than a centrifugal pump of the same speed.

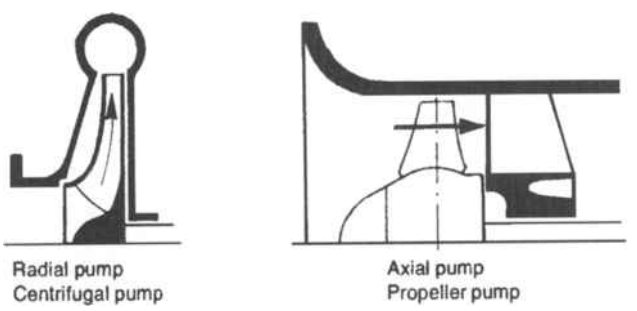

Figure 4.1 Examples of rotodynamic pumps

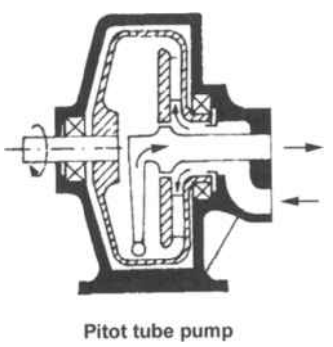

Figure 4.2 Example of special rotodynamic pumps

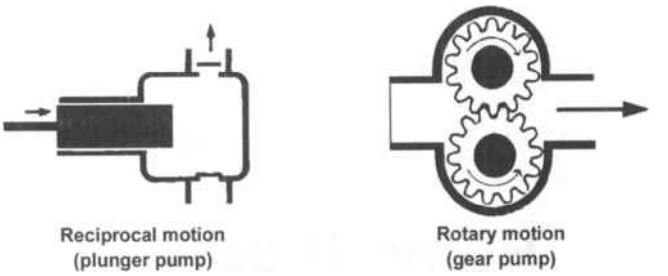

Figure 4.3 Examples of positive displacement pumps

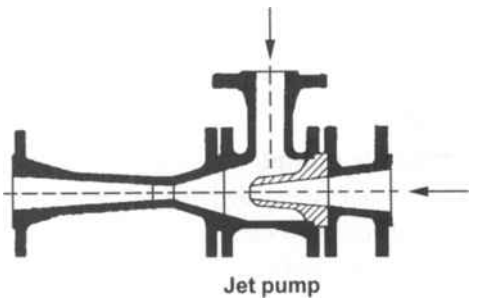

Figure 4.4 Example from the category Other pumps

Positive displacement pumps operate with enclosed liquid volumes which are forced forward in the direction of pumping, or squeezed and expelled into the pump outlet, see Figure 4.3. The oldest and perhaps the most typical positive displacement pump is the piston pump. A certain volume of liquid is pressed out of the cylinder for each stroke of the piston. The plunger pump is a variant of the piston pump. The piston pump belongs to the sub-category among the positive displacement pumps which operate by means of a reciprocating motion. The other sub-category comprises rotating or rotary positive displacement pumps. These include, for example, gear pumps where the liquid is transported in gaps between the gear teeth and where the difference in pressure is maintained by means of, among others, the seal afforded by the contact surface of the meshing gears. Other examples of rotary displacement pumps are screw pumps, vane pumps, hose pumps, etc.

Ejectors and air-lift pumps are typical examples of the main category termed "Other pump types". The static pressure in the nozzle of the jet pump falls when the speed of the motive fluid increases. The pumped liquid is sucked in and mixed with the motive fluid. The overall speed of the fluid flow is reduced and the static pressure is increased in the jet pump venturi (diffuser). Because jet pumps can handle liquids and gases they are useful for priming other pumps, see Figure 4.4. Jet pumps powered by steam or compressed air are also called ejectors.

Air-lift pumps function by reducing the effective density of the liquid by entraining large volumes of gas, usually compressed air. The liquid may be converted to foam if the surface tension is adequate. Atmospheric or gas pressure acting on the bulk liquid surface lifts the reduced density mixture above the stationary liquid. Performance is very variable and testing and tuning are usually required to achieve the desired results.

The performance of pumps, and above all rotodynamic pumps, is usually shown in the form of a curve. The main interest is the relationship between the volume of liquid transported per unit of time, the so-called volume flow or flow rate or just "flow" and the pressure increase or head rise of the pump. The different main categories have, in this respect, drastically different characteristics, see Figure 4.5.

The volume flow in positive displacement pumps is virtually independent of the pressure difference, i.e. the performance curve is almost a vertical line. The performance curve for the centrifugal pump shows increasing pressure rise with decreas-

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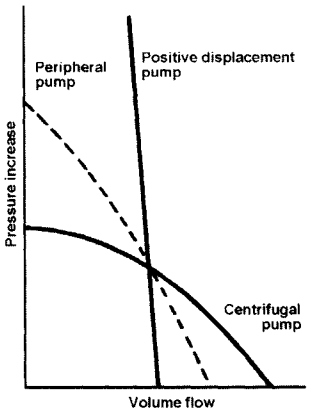

Figure 4.5 Pump curves for different pump types

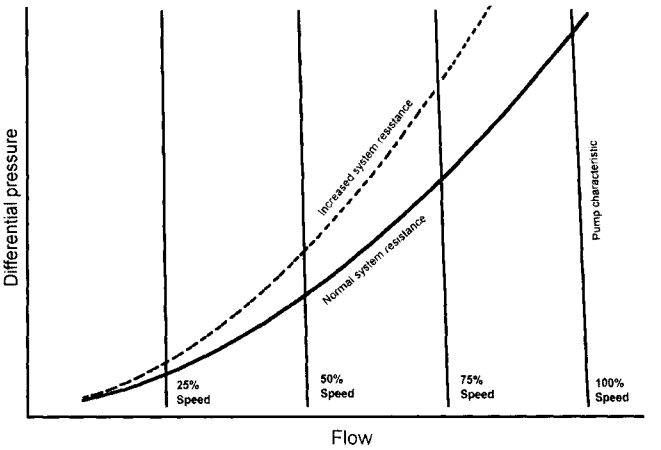

Figure 4.6 Positive displacement pump and system characteristics

ing volume flow. Peripheral pumps have a curve lying between these two groups of pumps, (see Chapter 1, Section 1.4.1). The flow in a pump is determined by the relationship of the specified pump pressure increase to the back-pressure, i.e. the resistance in the system, which is required in order to drive forward the flow.

The piping system has a characteristic curve, flow-differential pressure or head, which must be derived by the system designer. A pump can only operate where the pump characteristic intersects the system characteristic, see Figure 4.6. The pump characteristic can be modified by adjusting the pump speed. The system characteristic can be changed by opening or closing valves or by changing the levels in tanks. If everything remains constant the pump flow will be constant. However, the pump could wear and the pipework become partially blocked due to deposits or corrosion products. Operating conditions do tend to change with the passage of time.

If the liquid pressure in a pump suction is reduced to the vapour pressure of the liquid, the liquid evaporates. So-called cavitation occurs when the vapour bubbles return to liquid form in areas of higher pressure, this phenomenon causes intense local pressure spikes. Cavitation is not desirable since it reduces both pump performance and the operating life of the pump. The concept of NPSH/NPIP is employed to describe the risk of cavitation and states the necessary pressure or head measured above the vapour pressure of the liquid such that the effects of cavitation are limited or eliminated. For further details see Section 4.4.1.

4.2 Rotodynamic pumps

4.2.1 Rotodynamic pump theory

4.2.1.1 Differential head

In the past, pumps were used almost exclusively for transporting water from a lower to higher level, e.g. in a mine, water distribution or in an irrigation installation. Then it was practical to use the concept of delivery head (delivery height) as a criterion of the pumps performance ability. Despite the fact that today pumps are used to a large extent for purposes where the alteration in the elevation of the medium is of subsidiary importance, this concept of head still remains. It is useful because the ideal pump differential head is related to the speed of the vanes, velocity head $c^2/2$. In this Section the pump differential head will always be shown as ΔH and called differential head. In some publications it may be shown as $H_{t2\text{-}1}$ indicating the difference in total heads between suction and discharge. In pumping circles it is standard practice only to refer to "head" on the understanding that everyone knows that it is really differential head.

The process which takes place when a liquid is passing through the pump is isentropic, i.e. the exchange of heat between the pumped liquid and the environment is so small that it can be neglected and is more than compensated by extra heat generated by inefficiencies. An equal amount of mass flows through per unit of time in and out through the pump, the external leakage normally being negligible in comparison with the mass flow through the pump. Some pump designs have internal recirculation for thrust balancing which complicates the theory a little. Such a process is described by the Energy Equation (see also Chapter 3, Section 3.1.5) for steady flow calculated per unit of mass of pumped liquid and using the terms as given in Figure 4.7. Absolute flow velocity is designated by c in this Chapter in line with current practice for rotodynamic machinery.

$$l_t = u_3 - u_1 + \frac{p_3 - p_1}{\rho} + \frac{c_3^2 - c_1^2}{2} + g(h_3 - h_1) \qquad \text{Equ 4.1}$$

Put into words, the Energy Equation states the following: The mechanical work l_t, which is supplied via the pump shaft, causes: an increase of the internal energy (u), an increase in the static pressure (p/ρ), an increase in kinetic energy ($c^2/2$) and an increase in potential energy (gh), in the liquid.

It is desirable that as large as possible a part of the shaft work supplied results in an increase of static pressure in the liquid. In principle kinetic and potential energy can be completely converted into static pressure. It is more difficult to make use of the increase in the internal energy. It corresponds to an unnecessary and undesirable increase in temperature of the liquid and is considered as a loss. It is against this background that the pump's differential head ΔH is defined as that useful part of the change in state of the liquid measured in metres of liquid column.

$$\Delta H = \frac{p_3 - p_1}{\rho g} + \frac{c_3^2 - c_1^2}{2\,g} + h_3 - h_1 \qquad \text{Equ 4.2}$$

where:

p = static pressure (N/m^2)

c = absolute velocity (m/s)

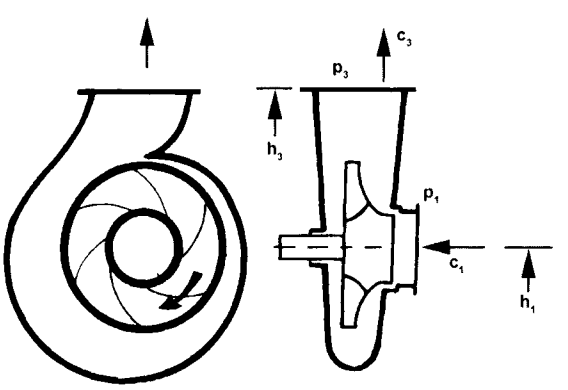

Figure 4.7 Pump terms

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h = potential head (m)

r = density (kg/m³)

g = gravitational acceleration (m/s²)

Note: The potential head of the inlet flow is taken at the centre-line of the impeller.

4.2.1.2 Losses and efficiency

Varying liquid properties can change the proportions of the power flow. Increased viscosity, for example, leads to higher impeller friction and blade losses. The shaft power P supplied is defined as the product of rotary moments and angular velocity at the pump's shaft coupling. The pump efficiency is designated by η.

$$\eta = \frac{P_u}{P} \qquad \text{Equ 4.3}$$

where:

P_u = useful work output (W)

P = power input (W)

The useful power (P_u) is thus the shaft power which remains from that supplied after all losses have been overcome. The useful power is applied to the pumped liquid and causes it to change state. The useful power is associated with the previously mentioned mechanical work according to the following relationship. (See Figure 4.8.)

$$P = \dot{m} \cdot l_t \qquad \text{Equ 4.4}$$

where:

$\dot{m}$ = the mass flow through the pump

$$P_u = \dot{m} \cdot g \cdot \Delta H = \rho \cdot Q \cdot g \cdot \Delta H \qquad \text{Equ 4.5}$$

where:

$\dot{m}$ = mass flow (kg/s)

ρ = density (kg/m³)

Q = volume flow (m³/s)

g = gravitational acceleration (m/s²)

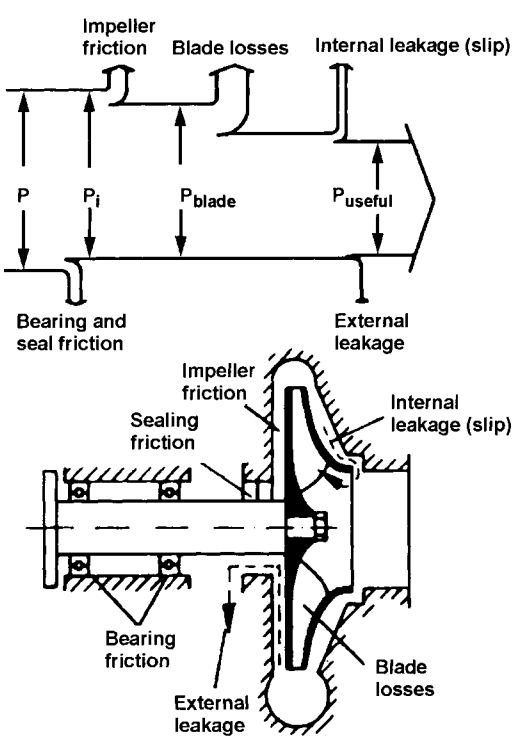

Figure 4.8 Power flow through a pump

ΔH = differential head (m)

The pump efficiency can be shown to be the product of other efficiencies concerned with specific losses within the pump.

$$\eta = \eta_m \cdot \eta_i \cdot \eta_c \cdot \eta_v \qquad \text{Equ 4.6}$$

A part of the shaft power is used in overcoming friction resistance in bearings and seals, P_{jm}. This loss of power does not generally affect the liquid but is transformed into heat which is given off to the environment. The pump mechanical efficiency is given by:

$$\eta_m = \frac{P - P_{jm}}{P} \qquad \text{Equ 4.7}$$

The impeller efficiency, η_i, is a measure of how much of the power available, ($P - P_{jm}$), is converted to kinetic energy in the liquid and how much power is lost in blade losses and disc friction.

The casing efficiency, η_c, is a measure of how much kinetic energy produced by the impeller, is recovered as static pressure rise compared to losses which appear as increased internal energy, temperature increase.

The volumetric efficiency, η_v, evaluates how much flow is delivered from the pump discharge compared to the flow through the impeller. External leakage must go through the impeller, and be worked on, before it can escape from the seal. Internal leakage, slip, allows a small quantity of liquid to recirculate around the impeller, reducing the effective pump flow.

The individual losses mentioned above are not constant but vary depending upon where the pump operates on its curve. At low flows/high heads, bearing friction can increase due to higher axial thrust and radial loads. The power conversion to kinetic energy in the impeller is influenced by the velocity triangles, see Figure 4.11. For a constant speed pump, the velocity triangles can only be correct for one flow rate, Best Efficiency Point (BEP). At other flows mismatches in the blade and liquid angles occur causing additional losses.

How the casing efficiency varies is dependant upon the type of casing design. Kinetic energy can be recovered as static pressure rise in two types of diffuser; vaned and vaneless. A vaned diffuser will suffer the same velocity triangle problems as the impeller with off-design flows. The vaneless diffuser or volute does not have blade angles to cause extra losses but does operate with flow paths away from the optimum in the specially shaped passages, again increasing losses. Internal leakage is affected by the pump differential head and the measures employed by the manufacturer to reduce it, such as wear rings, and wear in the pump.

A very important efficiency to consider, in today's increasingly energy conscious climate, is the overall efficiency, η_{gr}, often called the "wire-to-water" efficiency. This should be evaluated from the actual power consumed by the pump driver compared to the pump useful power. Overall efficiency takes into account any drive train systems between the driver and the pump and is the only valid efficiency to be used in Life Cycle Costs. Extra care should be exercised with electric motors controlled by variable frequency invertors. Invertors can inject harmonics into the mains supply which may cause serious errors in electrical power measurement.

When considering "wire-to-water" efficiency and Life Cycle Costs it is essential to consider the pump unit in a proper context. In the case of boiler feed pumps, applications where heat is added to the liquid after pressurisation, the normal pump efficiency is inappropriate. The heat added by the pump is not a loss but is a useful energy addition; the "wire-to-water" efficiency should be increased to include the pump heating effect. "Wire-to-water" efficiency can be maximised by matching the pump unit to the system requirements. Oversized pumps, throt-

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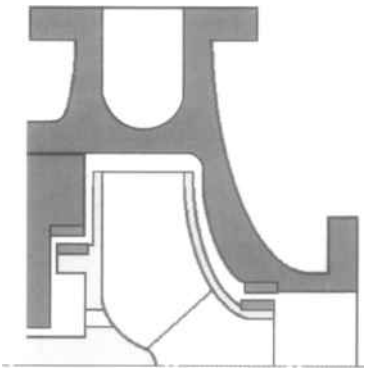

Figure 4.9 Wear rings on a centrifugal pump

tled to produce the correct flow, are very inefficient. As the power consumed by a pump costs much more than the initial purchase cost a small improvement in efficiency can result in a considerable reduction in Life Cycle Costs.

As mentioned earlier, wear rings are employed to reduce internal leakage slip. Figure 4.9 shows where rings are fitted on a centrifugal pump impeller and its casing. Sometimes rings are only fitted to the front of the impeller. Wear rings can be of various designs, see Figure 4.10. Wear rings are fitted to enclosed impellers, one on the front and one on the back. It is possible to avoid wear rings and just provide a tight clearance between the casing and the impeller, but adjustment for wear can be a problem. Wear rings are intended to be a replaceable spare part. Liquid leakage from higher pressure zones, after the impeller has imparted energy, to suction zones is reduced by imposing tight clearances in the leakage path. The actual radial clearance is strictly controlled.

Plain rings, shown on the left in Figure 4.10, are by far the most popular style. They are the easiest to manufacture and fit. Also, a plain ring does not restrict axial float. The rotor can move without rubbing on a ring. Stepped rings, shown in the centre, are fitted to some pumps, quite often at the insistence of the purchaser or pump user. Some pump users find a stepped ring provides better performance and longer life. The stepped ring concept can be taken further by introducing a radial flow portion to the ring combination. Stepped rings inhibit rotor axial movement and axial setting must be controlled accurately. Stepped rings also limit assembly possibilities.

Wear rings are primarily for internal leakage reduction, but do perform another important, secondary, function. Wear rings act as hydrodynamic bearings, bearings lubricated by the process liquid. The bearing function is quite useful in multistage pumps, when the wear rings effectively 'stiffen' the shaft; they reduce radial movement of vibration, for example. The material combination can be very important if rubbing is likely.

Rubbing is most likely to occur during start-up or shutdown, or during low speed operation when the hydrodynamic bearing effects are reduced. Wear ring material combinations are the result of 'trial and error' over the years, see Table 4.1. Newer materials, more expensive materials, can be adopted to solve problems such as very low viscosity or short term dry running. Alloys containing 75% or more nickel have proved effective. Modern engineering plastics, like PEEK, are proving very useful. The cost can seem exorbitant when compared to 'old-fashioned' metal.

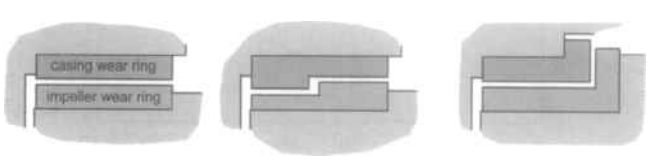

Figure 4.10 Common wear ring designs

Casing wear ring	Impeller wear ring
Cast iron	Cast iron
Ni-resist	Ni-resist
Bronze	Bronze
13Cr steel	13Cr steel
Stainless steel/Colmonoy 6	Stainless steel/Colmonoy 6

Table 4.1 Standard wear ring material combinations

Metal wear rings can be through-hardened, case-hardened or surface hardened. Laser hardening is used by some pump manufacturers. Most wear rings are ground to provide a very smooth bearing surface. Occasionally, wear rings are knurled to provide a rough surface. Micro-machining is a relatively new technique to improve bearing performance.

4.2.1.3 Euler's equation

Figure 4.11 shows a pump impeller with associated velocity triangle diagrams. The velocities u_1 and u_2 are the peripheral velocities of the impeller blade at inlet and outlet. The outlet peripheral velocity is commonly called the tip speed. Liquid enters the impeller at a velocity c_1 at a radius r_1. The liquid leaves the impeller at a radius r_2 with a velocity c_2. The design of the blade forces the relative velocity, i.e. the velocity which is experienced by an observer travelling with the movement of the blade, to alter its magnitude and direction from w_1 to w_2. Due to this the absolute velocity c_2 at the impeller outlet will also deviate from the flow velocity c_1 at entry. The tangential component of absolute velocity is designated by c_u, i.e. in the direction of the peripheral speed.

By applying the Law of Conservation of Momentum from fluid mechanics to the flow through the impeller, we obtain in the tangential direction the following

$$M_t = \dot{m}(r_2 \cdot c_{2u} - c_{1u}) \qquad \text{Equ 4.8}$$

Here M_t, is the rotary moment which the impeller must impart to the liquid, in order that the flow according to Figure 4.8 may be produced. The flow velocities in equation 4.8 are intended to be representative mean values for the liquid which is flowing through the impeller.

During the time Δt the pump shaft rotates through the angle $\Delta\phi$ at the same time as the mass Δm passes in and an equal mass Δm passes out through the impeller tip. If both sides of equation 4.8 are multiplied by $\Delta\phi$ and at the same time $\dot{m} = \Delta m/\Delta t$ is introduced, then

$$M_t \cdot \Delta\phi = \frac{\Delta\phi}{\Delta} \cdot \Delta m\,(r_2 \cdot c_{2u} - r_1 \cdot c_{1u}) \qquad \text{Equ 4.9}$$

but $M_t \cdot \Delta\phi/\Delta m$ is the work per unit mass which the rotary moment carries out whilst the impeller rotates through angle $\Delta\phi$. This work per unit mass is called blade work and is designated

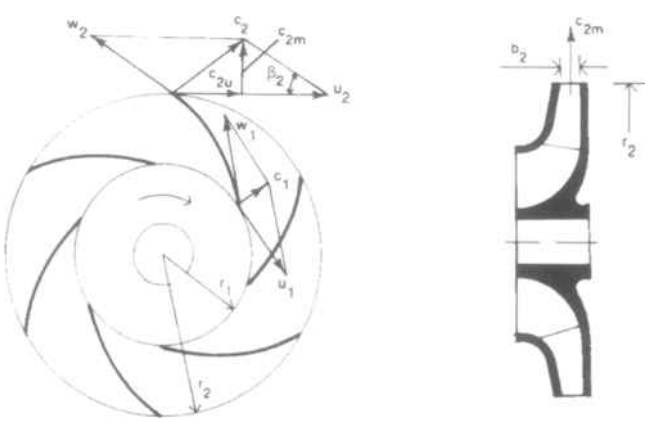

Figure 4.11 Impeller with velocity triangles at inlet and outlet

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l_b. Furthermore, the quotient $\Delta\phi/\Delta t$ is equal to the constant angular velocity ω of the pump impeller.

Thus:

$$l_b = \omega\left(r_2 \cdot c_{2u} - \phi \cdot c_{1u}\right) \qquad \text{Equ 4.10}$$

That part of the blade work which corresponds to the blade losses results in an increase of the internal energy of the liquid whilst the rest $(\eta_h \cdot l_b)$ provides a useful change of the state of the liquid $(g \cdot H)$. If then the peripheral speed $u = r_2 \cdot \omega$ is applied, we finally obtain:

$$\frac{g \cdot \Delta H}{\eta_h} = u_2 \cdot c_{2u} - u_1 \cdot c_{1u} \qquad \text{Equ 4.11}$$

where:

g	=	gravitational acceleration (m/s^2)
ΔH	=	differential head (m)
η_h	=	hydraulic efficiency
u	=	peripheral velocity (m/s)
c_u	=	tangential component of the absolute velocity (m/s)

Equation 4.11 is Euler's equation as it is normally written for pumps.

4.2.1.4 Pump curves

In accordance with Euler's equation differential head is dependent upon the size and direction of the velocity vectors and the size of the hydraulic losses. Both these factors are affected by, among others, the flow rate Q, which passes through the pump.

In many cases the tangential component of the absolute velocity entering the impeller is small, i.e. $c_{1u} \approx 0$. In such cases Euler's equation is simplified to:

$$\frac{g \cdot \Delta H}{\eta_h} = u_2 \cdot c_{2u} \qquad \text{Equ 4.12}$$

If the flow through the pump were free of losses $(\eta_h = 1)$, the ideal differential head, ΔH_i, would be given by the following equation:

$$\Delta H_i = \frac{u_2 \cdot c_{2u}}{g} \qquad \text{Equ 4.13}$$

ΔH_i is often called the Euler head. With reference to Figure 4.11

$$\Delta H_i = \frac{u_2}{g}\left(u_2 - \frac{c_{2m}}{\tan\beta_2}\right)$$

and

$$c_{2m} = \frac{Q}{2\pi \cdot r_2 \cdot b_2}$$

or

$$DH_i = \frac{u_2}{g}\left(u_2 - \frac{Q}{2p \cdot r_2 \cdot b_2 \cdot \tan\beta_2}\right) \qquad \text{Equ 4.14}$$

The angle β_2, in the velocity triangle is somewhat smaller than the blade angle at the impeller blade tip. This angular difference is called the deviation angle and is due to the blade's inability to control completely the relative flow of a real liquid as opposed to an ideal liquid with zero viscosity and density. The size of the deviation angle is primarily dependent upon the number of blades.

For a given pump, operating at a certain constant speed, the ideal differential head H_i decreases linearly according to equation 4.14 with increasing flow Q. The actual differential head H differs from H_i due to the hydraulic losses h_f. These, as in all other cases of flow, are dependent upon the inlet flow direction towards the body around which the liquid flows.

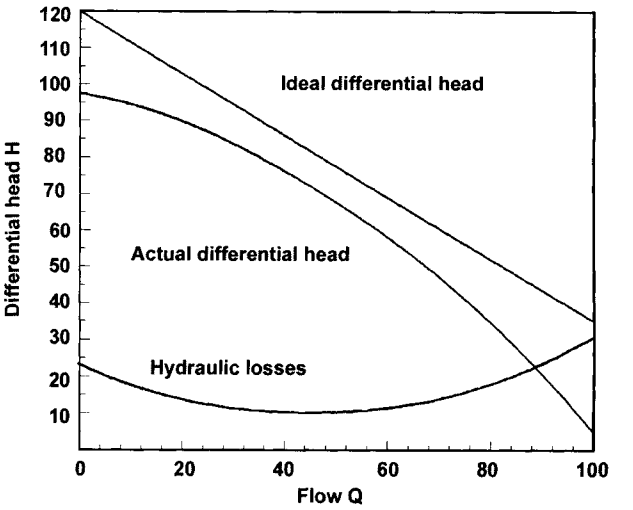

Figure 4.12 Centrifugal pump H-Q curve

In the case of pumps, the inlet flow direction towards for example the blades, varies with the flow rate. A certain inlet flow angle gives the most favourable flow and thus the smallest losses. Both higher and lower values of Q result in an increase of h_f. By subtracting h_f from H_i, the actual H-Q curve of the pump at constant speed is obtained, see Figure 4.12. The shape of the H-Q curve varies from one pump type to another and between different designs of the same pump types. Detailed design particulars, such as: blade profiles, blade angles, number of blades, area ratios and diffuser type, all affect the curve shape.

In addition to the pump's H-Q curve, the required shaft power P, the pump's efficiency η and NPSHR, as shown in Figure 4.13 are also usually given as a function of the flow in a pump characteristic curve. The shaft power is based on water, i.e. specific gravity = 1.0.

Variations in specific gravity will affect the power, and the discharge pressure even though the differential head is constant. The pump can, in principle, operate at any point whatsoever along the H-Q curve. The position of the operating point in an actual case is determined by the characteristics of the system, suction plus discharge, to which the pump is connected. The art of good pump selection is to operate as close as possible to the Best Efficiency Point (BEP); also some users specify the duty point must be to the left of BEP.

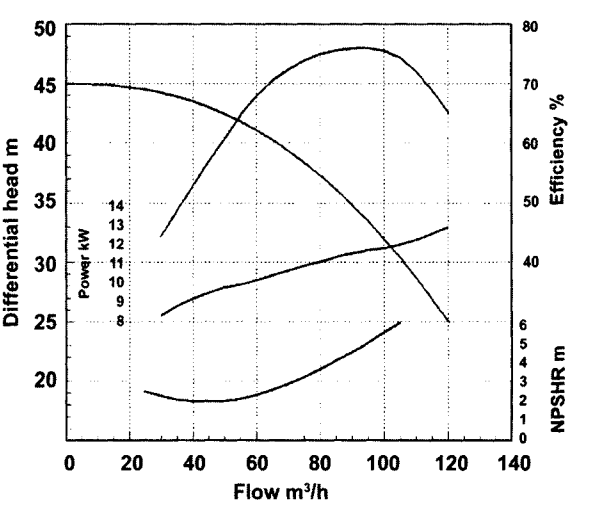

Figure 4.13 Constant speed, fixed diameter pump curve

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4.2.1.5 Specific speed

An important parameter which is frequently used to describe different pump types and to characterise a duty point is specific speed N_s, defined by the following relationship:

$$N_s = n \cdot \frac{Q_{BEP}^{0.5}}{H_{BEP}^{0.75}} \qquad \text{Equ 4.15}$$

where:

N_s = specific speed (dimensionless)

n = shaft speed (rpm)

Q_{BEP} = pump flow at best efficiency point (m^3/h)

H_{BEP} = pump ΔH at best efficiency point (m)

Note that N_s is not dimensionless and therefore has different numerical values in different systems of units. N_s can be defined in words as the mechanical speed of a pump identical with the one under consideration, and which with identical velocity vector diagrams gives a flow of 1 m^3/h at a differential head of 1m. All identically shaped pumps have therefore the same specific speed independent of their size.

The specific speed is used, among others, for characterising the shape of rotodynamic pump impellers, see Figure 4.14. Drawing on experience gained in this field, it is generally possible to estimate which impeller shape will, under normal circumstances, give the best results for a given speed, flow and differential head. Figure 4.13 illustrates this relationship. The axial flow pump has shown itself to be the most suitable type in the case of, for example, a large flow and a small differential head, i.e. in the case of high specific speed. Axial flow pumps are therefore said to have high specific speeds.

There is another form of specific speed in popular use, Suction specific speed, N_{ss}. Suction specific speed is evaluated using the same equation as specific speed but NPSHR is used in place of differential head. N_{ss} is an indication of how good the suction performance of a pump is. Some users specify a limiting range, 175 to 215, of N_{ss} in an attempt to limit cavitation problems.

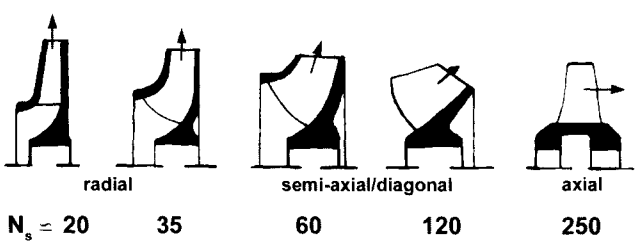

Figure 4.14 Pump impellers having different specific speeds

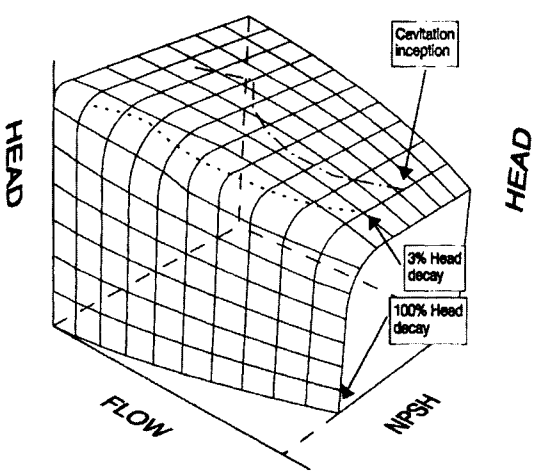

Figure 4.15 Pump performance surface

Cavitation, especially when operating well away from BEP, is a very complex phenomenon and specifying any N_{ss} value will not necessarily solve or help the problem. Other problems associated with rotodynamic pumps operating well away from the BEP are surge and internal recirculation. Both of these problems are associated with low flows. Manufacturers take great care to avoid surge and recirculation problems by specifying minimum flows for pumps. It is up to the user to correctly assess the full operating regime and communicate this to the manufacturer.

A very interesting and educational paper was presented by R. Palgrave, some years ago, on the relationship and effects of both specific speed parameters. The paper, indicates the alternatives available to the pump designer in response to changing operating conditions and customer restraints. The paper shows pump performance as a 3D surface, which connects flow, head and NPSH, rather than a conventional 2D curve and is shown in Figure 4.15.

4.2.1.6 Similarity

Under certain conditions, the performance of a pump can be recalculated in a specially simple manner which is valid for all sizes of pump at different operational speeds. These conditions are as follows:

- The pumps compared should be identical or geometrically similar
- The pumps compared should operate at similar operational points, i.e. with geometrically similar velocity diagrams
- Any difference in efficiency should be negligible

In a comparison between two pumps I and II, which fulfil these conditions, the following relationships between their respective speeds n_I and n_{II}, (dimensions expressed as the impeller diameters D_I and D_{II},) and their performance, will apply:

- For the flows

$$\frac{Q_I}{Q_{II}} = \frac{n_I \cdot D_I}{n_{II} \cdot D_{II}}$$

- For the differential heads

$$\frac{H_I}{H_{II}} = \frac{n_I^2 \cdot D_I^2}{n_{II}^2 \cdot D_{II}^2}$$

- For the power requirements

$$\frac{P_I}{P_{II}} = \frac{n_I^3 \cdot D_I^3}{n_{II}^3 \cdot D_{II}^3} \qquad \text{Equ 4.16}$$

The Affinity Laws which apply to a pump with a fixed impeller diameter, form a special case of the general similarity relationships. When $D_I = D_{II}$, then

$$Q \propto n$$

$$H \propto n^2$$

$$P \propto n^3 \qquad \text{Equ 4.17}$$

When the principles of similarity are used in practice, the variation in efficiency between the pumps compared is ignored except when calculating power. Similar pumps have the same blade angles. Geometrically similar velocity diagrams imply the same flow angles. The inlet flow direction, which is so important for the hydraulic losses, is therefore unaltered in the cases compared. Against this background a constant hydraulic efficiency would seem to be a reasonable approximation. In practice the efficiency will improve slightly as speed increases. In the case of very large pumps which are tested by means of model tests in the laboratory, a somewhat higher efficiency can be expected for the full-scale pump than that which is mea-

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sured during the model test. Parasitic losses tend to increase as size reduces.

Similarity is very useful to pump manufacturers. Similarity allows a pump curve to be adjusted for changes in pump speed and small changes of impeller diameter. Similarity is not quite so useful for pump users. Similarity does not predict changes in operating conditions. The pump curve can be adjusted then the system curve must be added. A pump can only operate at the intersection of the pump characteristic and the system characteristic. The similarity equations apply to pumps — not pumps in systems!

4.2.1.7 Axial thrust and radial forces

The pump impeller works best if the impeller tip width is aligned centrally with the throat of the diffuser. The pump shaft is adjusted during assembly to accomplish this. The unbalanced axial thrust (axial force) on the pump shaft must be absorbed by a thrust bearing or balanced out with a special arrangement. A special balance disc or drum, however, always requires a certain amount of driving power and internal leakage and therefore reduces the overall efficiency of the pump. It is thus advantageous if the axial force can be limited directly by means of the pump impeller design. This is not always possible and some large pumps have a balance disc/drum to reduce the axial thrust and a thrust bearing to locate the shaft. Balance discs and drums are always designed to leave a residual thrust which must always be in the same direction to avoid thrust bearing problems. (See Figure 4.16.)

The application of the Momentum equation, equation 3.7, in Chapter 3, to a control volume, precisely that which encloses the impeller, gives the following, using the terms as shown in Figure 4.17.

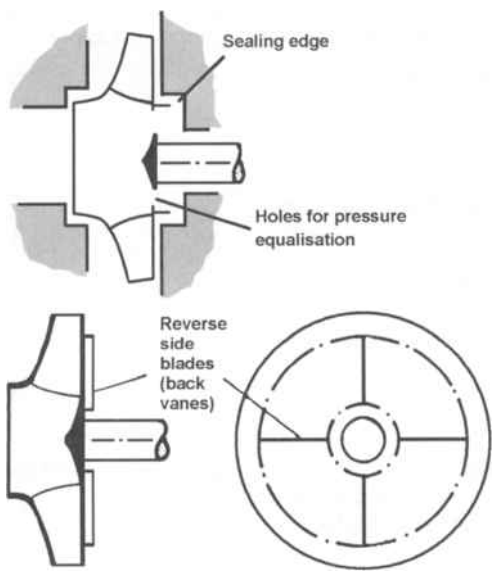

Figure 4.16 Methods for limiting the axial thrust

$$F_{ax} = \int_{r_a}^{r_s} p_b \cdot 2\pi \cdot r \cdot dr + \int_{r_s}^{r_2} p_b \; 2\pi \cdot r \cdot dr - \int_{r_s}^{r_2} p_f \cdot 2\pi \cdot r \cdot dr - p_s \cdot p \cdot r_s^2 + \dot{m} \cdot c_{2m} \cdot \sin\beta - \dot{m} \cdot c_s \qquad \text{Equ 4.18}$$

If the pressure on the front and back face of the impeller for $r < r_s$ is assumed to be equal ($p_b = p_f$) and if the angle β is small ($\sin\beta = 0$) the above expression can be simplified to

$$F_{ax} = \int_{r_a}^{r_s} p_b \cdot 2\pi \cdot r \cdot dr - p_s \cdot \pi \cdot r_s^2 - \dot{m} \cdot c_s \qquad \text{Equ 4.19}$$

In order to limit F_{ax} it is normal procedure to provide the back wall of the impeller with an extra sealing edge at the same diameter as r_s, back wear ring, and at the same time to bore holes through the wall close to the shaft for pressure equalisation. Then

$$F_{ax} \; - p_s \cdot \pi \cdot r_a^2 - \dot{m} \cdot c_s \qquad \text{Equ 4.20}$$

As well as the axial force from the impeller, the shaft is also subjected to forces from seals and bearings as well as the atmospheric pressure at the end of the shaft. The bearing force is

$$F_{ax\,bearing} = (p_a - p_s) \cdot \pi \cdot r_a^2 - \dot{m} \cdot c_s + F_{shaft\,seal} \qquad \text{Equ 4.21}$$

In general the forces generated by seals and atmospheric pressure on the open shaft end are very small. Another way of limiting the axial thrust is to provide the impeller with blades on the reverse side. This causes the rotation of the liquid on the reverse side to be greater than that achieved by friction on the front side. Again see Figure 4.16. The increased rotation therefore causes lower pressure for $r < r_2$ and also therefore a reduction in the actual force. In the case of multi-stage pumps or double suction impellers, low axial thrust can be obtained by having impellers facing in opposite directions.

Considerable radial forces occur during partial loading in all pumps equipped with volute casings. The design of such pumps seeks to achieve an even pressure distribution around the impeller. This condition determines the shape of the volute casing. During part load, however, the pressure around the periphery of the impeller will vary. The radial forces on the impeller are taken up by the bearing via the shaft. The diameter of the shaft and the distance from the bearings determines the deflection.

Since the pressure varies around the periphery of the impeller the flow in the individual blade passages will also vary as the impeller rotates. In simple theory the liquid flow is considered as constant velocity across the peripheral section of a blade passage. In reality the flow may be in two or three distinct zones with different velocity profiles.

The size of the radial force can be estimated by means of the following formula:

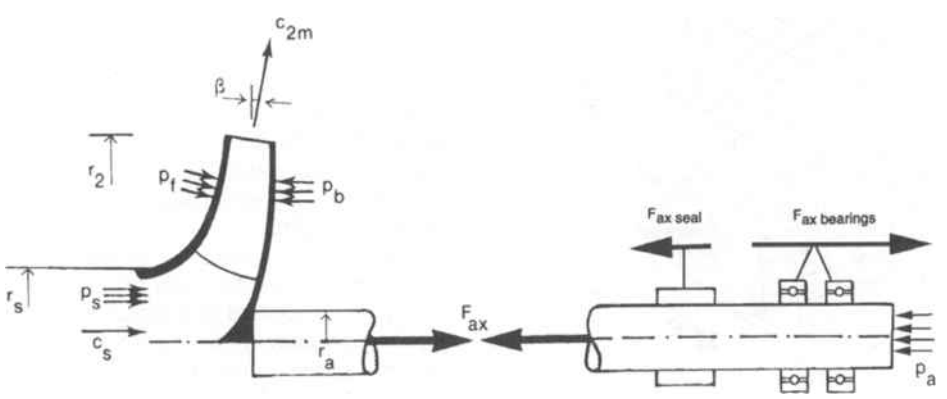

Figure 4.17 Illustration of axial thrust

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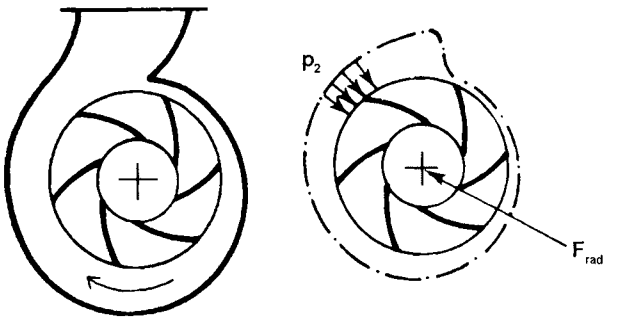

Figure 4.18 Illustration of radial forces on the impeller in the case of reduced flow ($Q < Q_{BEP}$)

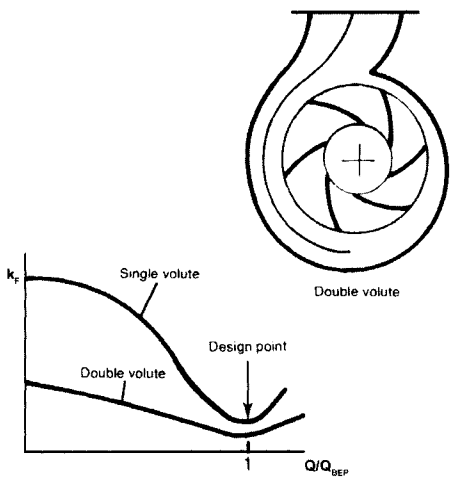

Figure 4.19 Examples of the effect of the position of the operating point and the type of the volute casing on the parameter k_F

$$F_{rad} = k_F \cdot D_2 \cdot b_2 \cdot \rho \cdot g \cdot \Delta H \qquad \text{Equ 4.22}$$

where:

F_{rad}	=	radial force (N)
k_F	=	empirical constant (dimensionless)
D_2	=	diameter of impeller (m)
b_2	=	tip width (m)
ρ	=	density of liquid (kg/m^3)
g	=	gravitational acceleration (m/s^2)
ΔH	=	differential head m)

The parameter k_F assumes different values for different designs of pumps and also varies greatly with the volume flow through the pump. Figure 4.18 illustrates this.

For a centrifugal pump with a single volute, k_F can have a value up to 0.4 at zero discharge (Q=0).

An effective method of limiting the size of the radial force is to provide a double volute, see Figure 4.19.

The radial force is greatest at Q=0, the shut-off head. The radial force causes a deflection of the pump shaft and subjects the shaft to rotary fatigue. The condition that the maximum shaft deflection at the zero discharge should be less than 0.05 mm at the shaft seal is often used as the guiding value in deciding the dimensions of the shaft.

The size of the axial thrust and radial force is a determining factor in the design and construction of the pump (clearances, bearings, bearing arrangement, etc.). They are also often the primary cause of breakdowns. Note that both the axial thrust and the radial force increase with partial load and are greatest at zero discharge. Pumps with front and back wear rings on each impeller have much stiffer shafts than simple theory predicts. Each wear ring acts as a hydrodynamic bearing, to some extent, and provides support for the shaft.

Large pumps, and high differential head pumps, often have restrictions regarding low flow operation. The manufacturer may have stated the pump should not be operated continuously at less than 25% flow. Low flow operation is restricted to reduce heat build up in the pump; it obviously helps with axial thrust and radial load.

4.2.2 Rotodynamic pump curves

4.2.2.1 Presentation of data

In addition to the relationship between head and flow (H-Q) the complete data set for the pump also comprises power input, efficiency and NPSHr. Figure 4.20 shows an example of a complete pump characteristic. This diagram also shows performance curves for different diameters of impeller, i.e. the different performances which can be attained for one and the same pump by machining the impeller. With moderate adjustments of the impeller diameter the NPSH curve is not affected and the efficiency curve is only affected to an insignificant degree.

Rotodynamic pumps are usually designed to be driven directly by standard squirrel-cage electric motors. This philosophy means that most pumps run at one speed out of a choice of eight.

Number of poles	50 Hz	60 Hz
2 pole	2900 rpm	3500 rpm
4 pole	1450 rpm	1750 rpm
6 pole	960 rpm	1160 rpm
8 pole	730 rpm	880 rpm

Table 4.2 Approximate full load speeds of electric motors

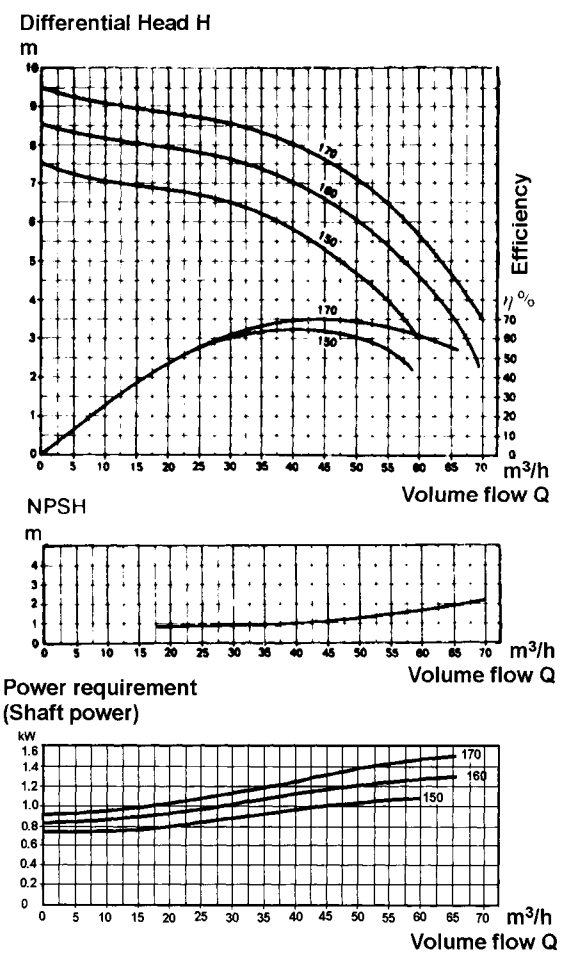

Figure 4.20 Example of performance curve for a centrifugal pump with a speed of 1450 rpm on water (Numbers 150-170 refer to the impeller diameter in mm)

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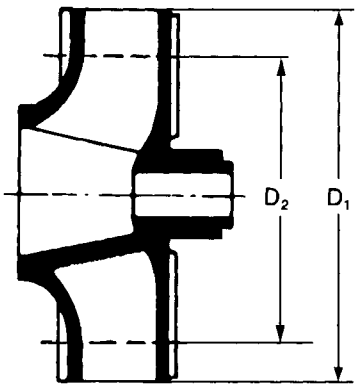

Figure 4.21 Impeller with diameter reduced from D_1 to D_2

Motors are available with more than eight poles but these are not popular and are costly. The actual full load speed of the motor is dependant upon its size. Large motors are more efficient and have less slip and therefore run closer to the synchronous speed. From Table 4.2 it can be seen that there is a speed ratio of about two between 2-pole motors and 4-pole motors.

For a pump designed at 2900 rpm to run at 1450 rpm it would produce 50% flow and 25% differential head. Pump manufacturers needed a cost-effective mechanism to increase the operating range of a pump without changing motors or speeds with gearboxes or V-belts. Reducing the impeller diameter, from the maximum possible, is a simple and very effective solution. The impeller can be reduced to about 70% on most designs.

In the absence of pump performance data for different diameters, the modification in performance in the case of reduction in the diameter from D_1 to D_2, according to Figure 4.21, can be estimated according to the following rule:

$$\frac{H_2}{H_1} = \frac{D_2^2}{D_1^2} \quad \text{and} \quad \frac{Q_2}{Q_1} = \frac{D_2}{D_1} \qquad \text{Equ 4.23}$$

A new approximate H-Q curve is obtained by recalculating sufficient points to produce a smooth curve.

The efficiency is sometimes indicated by iso-efficiency lines on the H-Q diagram as shown in Figure 4.22.

When a pump is designed to be capable of accepting different types and sizes of driver, the pump characteristic will show the efficiency of the pump alone. If the characteristic shows an overall efficiency, including driver and any transmission losses, it will be stated clearly.

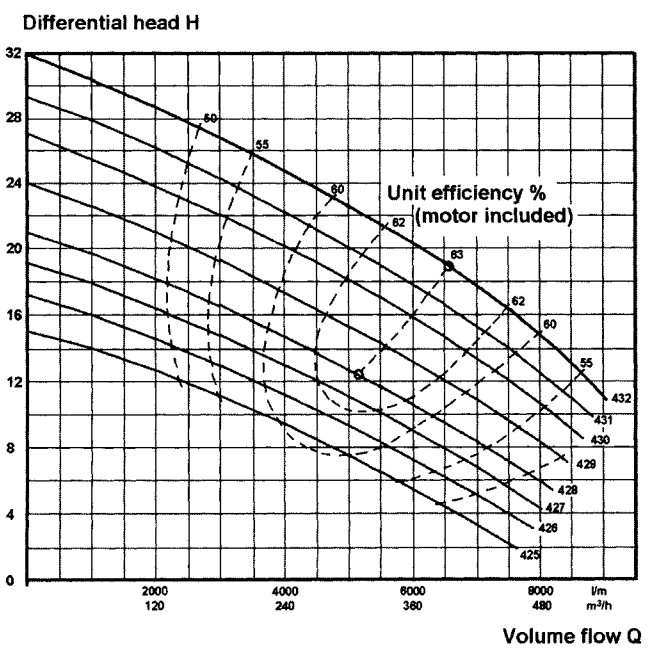

Figure 4.22 H-Q diagram with iso-efficiency lines (Numbers on the extreme right of each H-Q curve are the different impeller diameters)

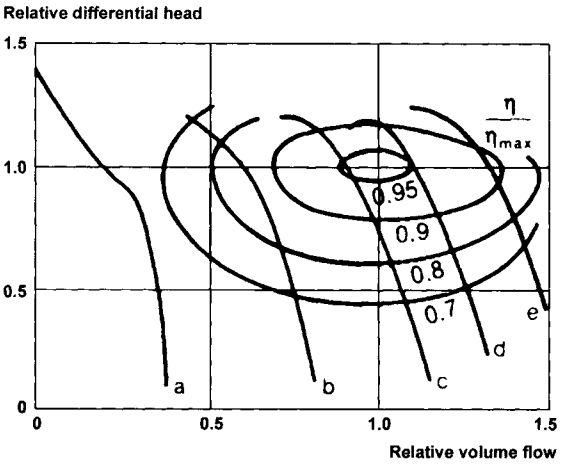

Figure 4.23 Pump curves for an axial flow pump with adjustable impeller blades. The curves a to e designate different blade settings. The performance is expressed relative to Best Efficiency Point (η_{max})

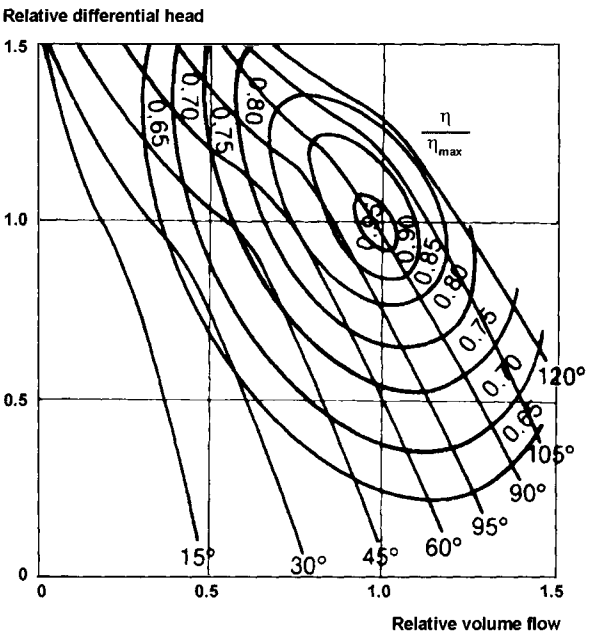

Figure 4.24 Pump curves for mixed flow pump with adjustable inlet guide vanes (The angular values relate to the angle setting of the guide vanes)

In the case of axial or mixed flow pumps, different characteristics can be achieved through adjustment of the angular position of the impeller blades, see Figure 4.23. Similarly the performance can be modified by adjusting stationary inlet guide vanes, see Figure 4.24.

If the pump speed is varied then performance changes can also be expressed as a series of curves in the H-Q and the P-Q diagrams. For example, an existing pump curve with H-Q and P-Q diagrams can be recalculated from a maximum speed to some other speed by recalculating 4 or 5 points on the maximum curves, one after the other, to the new speed; and with the help of the affinity laws and the transmission efficiencies for the actual speed converter. Figure 4.25 shows an example of such recalculated pump performances. A more detailed treatment of speed regulated pumps is given in Chapter 6.

The pump curves described so far have related to one and the same size pump. In order to be able to display data for many sizes of pump simultaneously, range charts are used. Logarithmic scales may be used on either or both axes to show the individual performance range with some accuracy, see Figure 4.26. Diagrams of this type are used both in order to be able to find a suitable size of pump quickly, and to systematise classifi-

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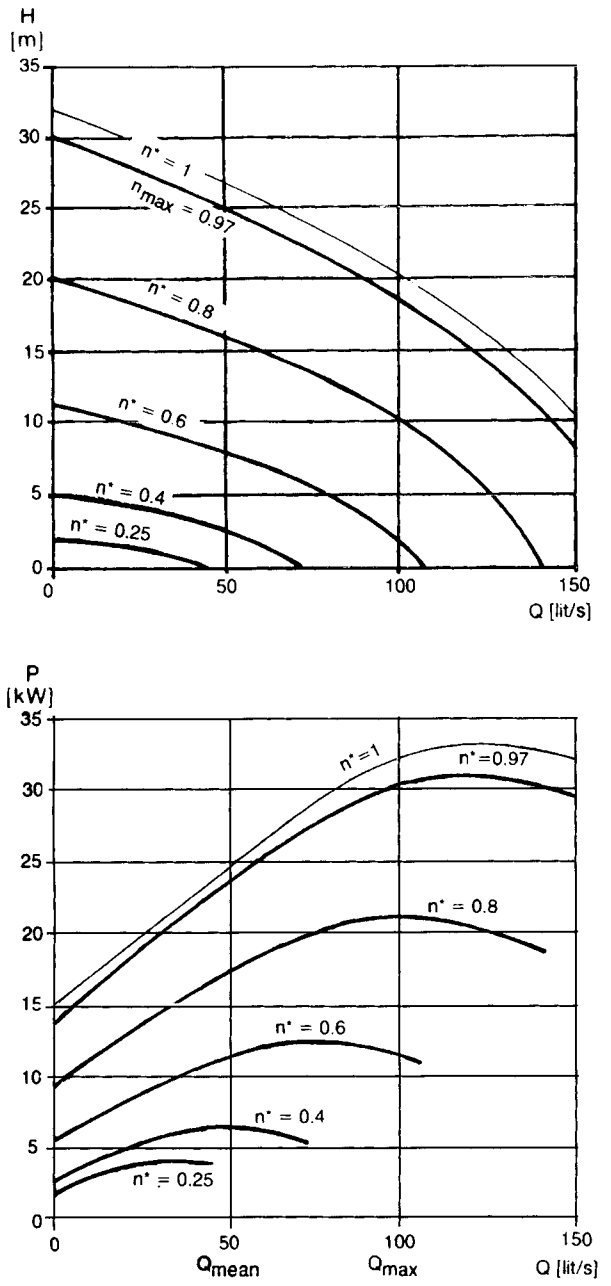

Figure 4.25 Example of performance curves for a variable speed pump. The power requirement relates to a pump with fluid coupling. The speed is expressed relative to the full speed of a direct driven pump. Due to slip in the hydraulic coupling at the full load, $n^*_{max} = 0.97$ which gives somewhat lower performance than for corresponding direct drive pumps

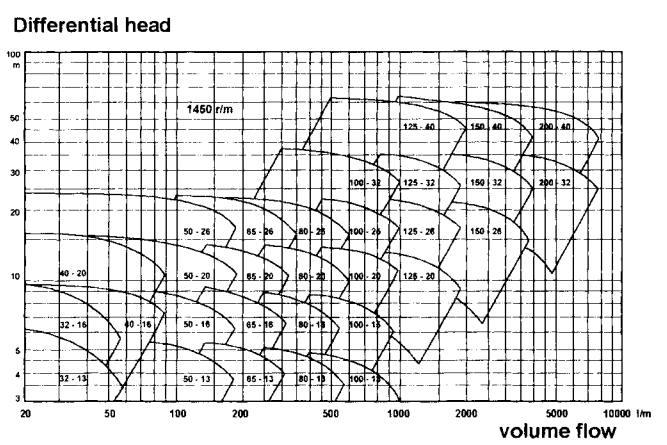

Figure 4.26 Range chart for small single stage pumps. (The first number of each designation relates to the discharge dimension in mm and the second number to the maximum impeller diameter in cm)

cation into different sizes. Due to the use of different pump impeller diameters or blade angles, one pump size covers a "window" in the diagram. Each pump covers a flow range of approximately 2:1 and a head range of 1:6. In order to cover varying needs for volume flow and differential head, 20 to 60 different sizes of pump can be required in a range to provide good coverage. It is very useful to indicate the BEP as a point on the maximum and minimum impeller curves. This shows very quickly how good a selection will be.

Good range design for a specific type of pump is very complicated and many factors must be considered. Problems arise because small pumps cost more to manufacture than large pumps. Good range coverage is achieved with centrifugal pumps when pumps are spaced on a geometric power scale. The step size should lie between 1.2 and 1.75. Experience has shown that because of the small pump cost problem, small pump design should not be compromised by interchangeability considerations but designed specifically for that size.

A rotodynamic pump's H-Q curve is stated as being stable or unstable depending upon whether the differential head is constantly rising or not when flow decreases, Figure 4.27. The differential head when Q = 0 is called "shut-off head" or "closed valve head" and in the case of a centrifugal pump with a stable pump curve is the highest pressure the pump can produce. The unstable part of the curve can cause difficulties due to the fact that the point of interception with the system curve is not clearly definable in some cases. Unstable pump curves are therefore undesirable and are usually avoided when the friction losses in the system are small and when a number of pumps are operating in parallel.

Depending upon the slope of the H-Q curve, a distinction is made in theoretical considerations between steep and flat curves. As a measure of this slope, the equation H_{max}/H_O, is evaluated for the differential head at optimum BEP and closed valve, as shown in Figure 4.28.

When drawing a H-Q curve the choice of scales can cause the same pump curve to appear to be flat or steep. Choice of an operating point to the right or left of the flow at best efficiency, Q_o in Figure 4.28, determines in practice if the curve, together with a piping system will function as a steep or a flat pump curve.

Parallel pump operation can be very difficult, especially when the system curve is mostly static pressure and there is little pipe friction, such as in boiler feed pumps. Experienced users specify a minimum value, percentage head, for the slope of the H-Q curve between the duty point and closed valve.

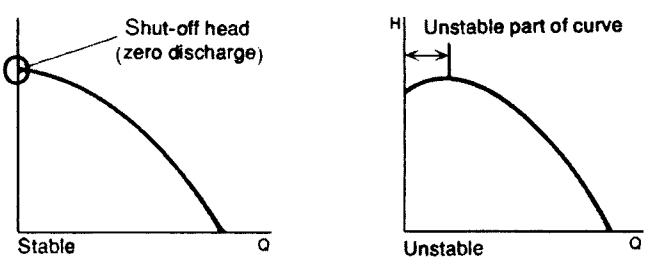

Figure 4.27 Stable and unstable pump curves

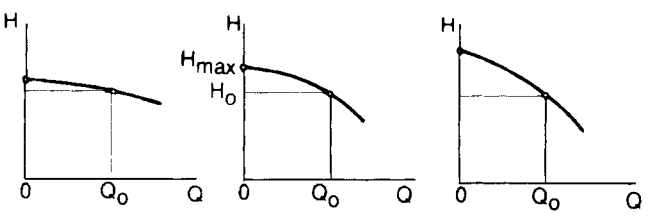

Figure 4.28 Flat and steep pump curves for centrifugal pumps

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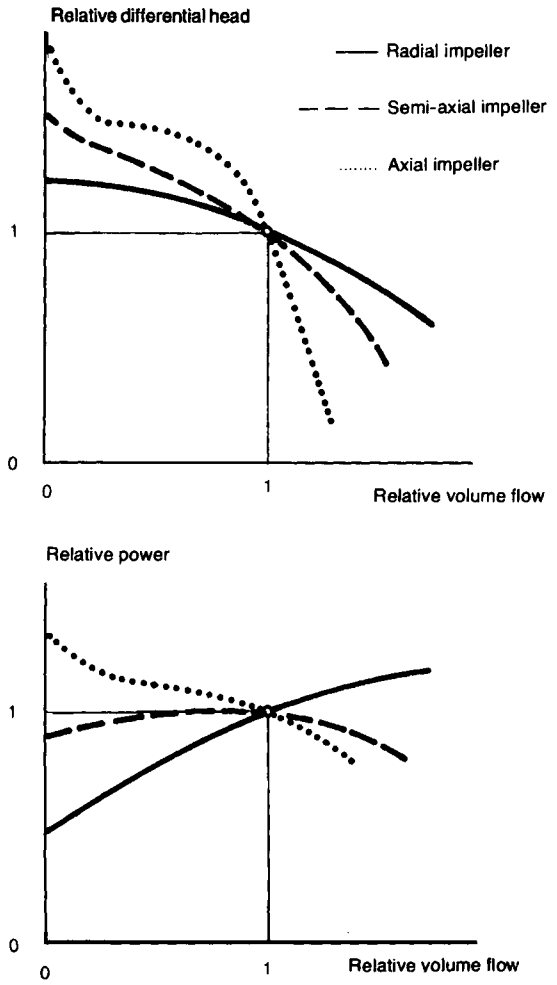

Figure 4.29 Rotodynamic pump curves relative to the BEP

The many varieties of rotodynamic pump — centrifugal pumps, mixed-flow pumps, axial pumps; have performance curves which differ greatly in appearance as shown in Figure 4.29. An increased specific speed gives an increasingly steep H-Q curve whilst the power curve alters from rising to decreasing with the flow.

The shape of the power curve is, together with variations in the flow, the determining factor for the size of the driver. In the case of axial pumps, see Figure 4.29, the greatest power requirement is when the flow is zero, which can mean that the pump's starting conditions must be adapted to this fact. The power requirements which are given in pump catalogues, unless otherwise stated, refer to liquids comparable with cold water, i.e. with a density of 1000 kg/m^3 and a viscosity of 1 cP approximately.

In the case of liquids having different densities and viscosities from that of water, the use of the unit "m" for differential head can cause misunderstandings. "m" stands for "metres of liquid column". If reference is made to "metres of water column" it is usually written "m H_2O" to avoid confusion. The reason for this confusion is that pumps give the same differential head in meters liquid column (m), irrespective of the density of the liquid. The pump's power requirement, and discharge pressure, is on the other hand proportional to the density. For densities different from that of water the power requirements and discharge pressure quoted in the purchase order always relate to the liquid stated. When pumping liquids which differ from water, the manufacturer should state how the pump will perform, on test, with water.

The standards for acceptance testing of rotodynamic pump performances is covered by various current national and international standards, for further details see Chapter 17, Section 17.1.9 and Chapter 12. The manufacturers' guarantees relate

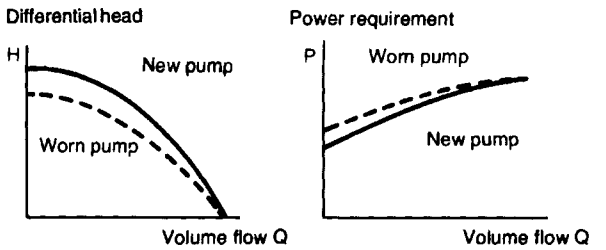

Figure 4.30 The effect on performance of wear in seals and clearances

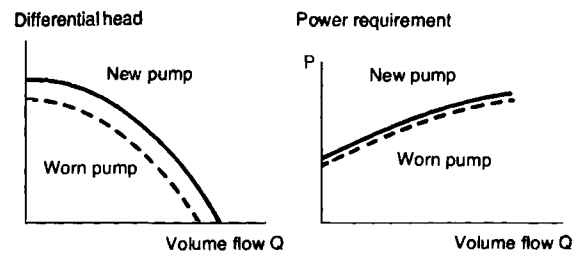

Figure 4.31 The effect on performance of wear on pump blades

to an installation designed and built to recognised standards and sound engineering practice. An unsuitable design of, above all, the inlet pipe in an installation can reduce the pump performance and cause severe operational problems.

A pump's performance falls off with wear of the pump parts. This wear occurs both at the internal and external seals and at clearances, affecting the internal and external leakage, see Figure 4.30; and at the blades, decreasing the blade work, see Figure 4.31. It is very difficult to give a limit for the extent of wear but reductions in delivery head of the order of 10% during the pumping of clean liquids are, however, possible.

The performance of pumps falls off rapidly with increasing viscosity of the pumped liquid. This reduction expresses itself in such a manner that the H-Q curve falls but shut-off head is retained. The power requirement rises considerably due primarily

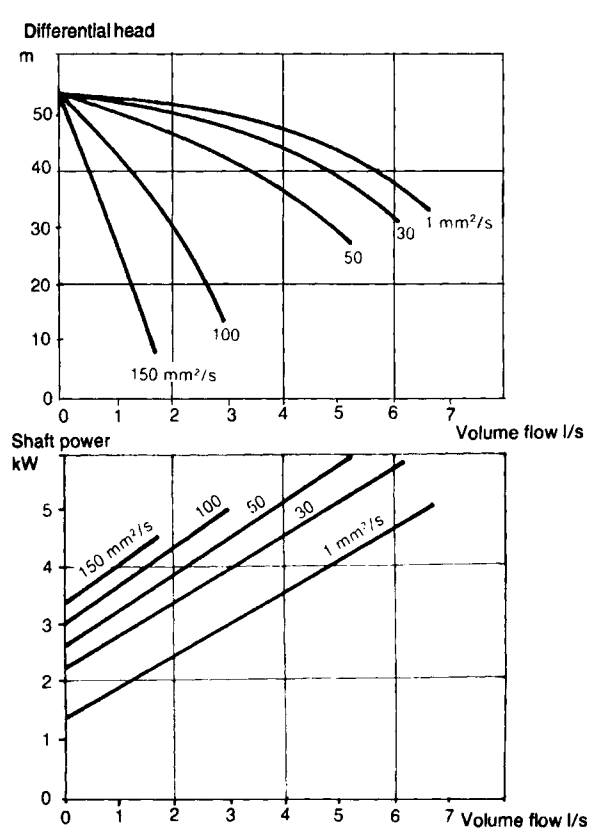

Figure 4.32 The effect of viscosity on a small centrifugal pump N = 11, discharge connection 50 mm. Max efficiency falls from approx 50% to approx 5%

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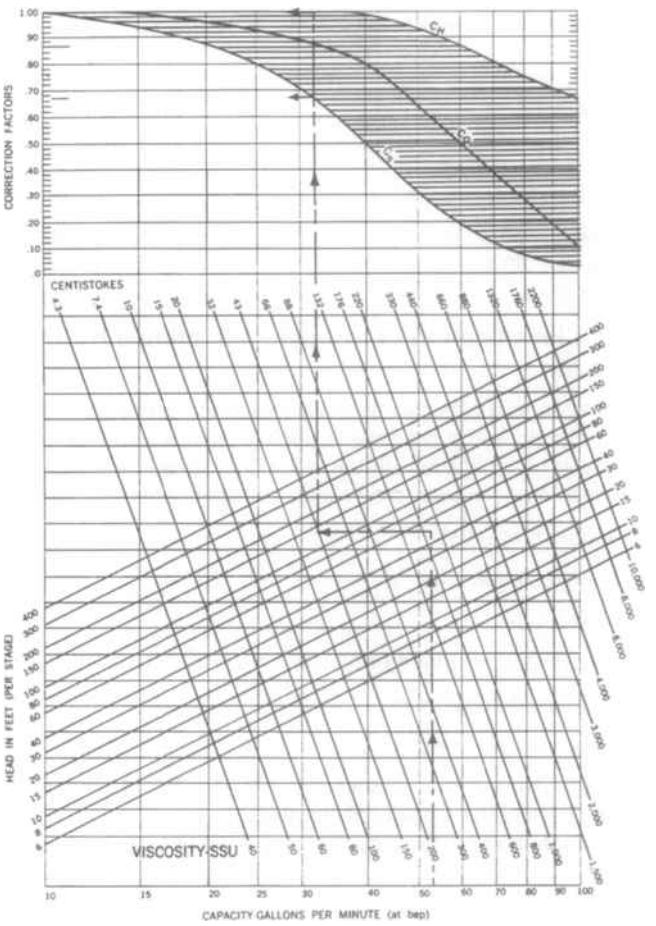

Figure 4.33 Viscosity corrections for small centrifugal pumps

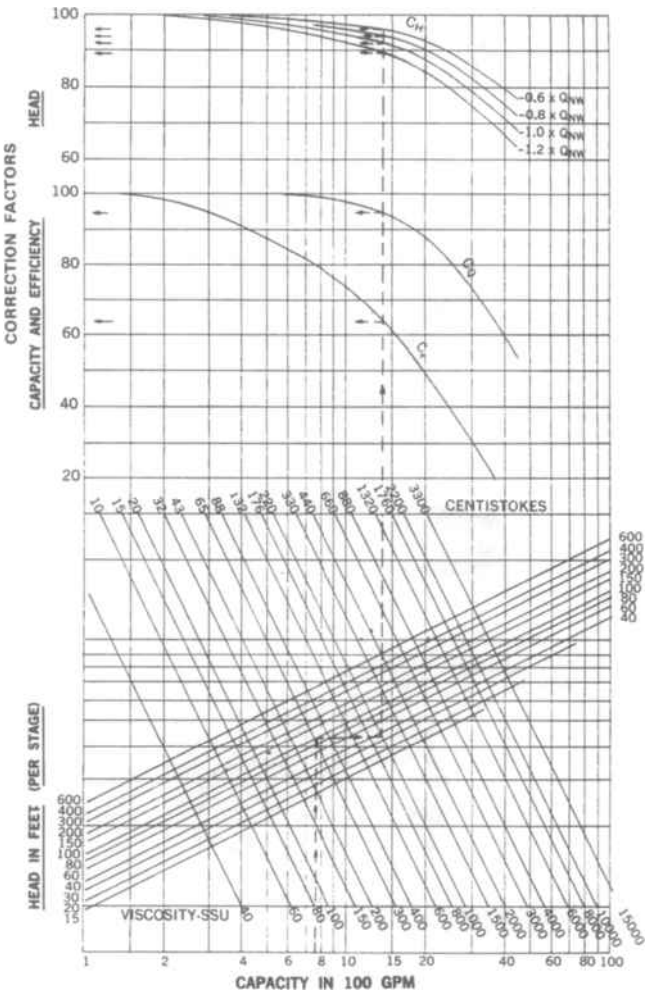

Figure 4.34 Viscosity corrections for large centrifugal pumps

to the increase of impeller friction. Figure 4.32 shows typical examples of the reduction in performance for smaller centrifugal pumps.

Figures 4.33 and 4.34 show correction factors, in USgpm and feet, for large and small centrifugal pumps. These graphs are based on testing carried out by Stepanoff of the former Ingersoll-Rand and are reproduced by permission of Flowserve Corporation. For further details see Chapter 15 on pump selection. PD ISO/TR 17766:2005 uses the Stepanoff data to produce a mathematical correction factor method. PD ISO/TR 17766:2005 gives performance corrections for all worldwide designs of centrifugal pumps of conventional design, in the normal operating range, with open or closed impellers, single or double suction, pumping Newtonian fluids.

4.2.3 Classification of rotodynamic pumps by hydraulic design

4.2.3.1 Impeller

The impeller, together with the casing, constitute the most critical parts in a rotodynamic pump. The shape of the impeller is determined by a combination of factors: flow, differential head and speed. These three parameters are used to define the specific speed in accordance with equation 4.15, in Section 4.2.1.5. In principle all pumps with the same specific speed can be geometrically similar. The variation, in practice, is due to different design techniques and manufacturing methods for small and large impellers and also the effect of surface roughness.

Experience teaches the hydraulic designer how the impeller should be shaped in order to obtain the greatest efficiency and suitable curve shape. The pump is named according to the flow path through the impeller, such as: centrifugal, mixed flow and axial, see Figure 4.35. There are an unlimited number of intermediate impeller shapes to those shown in order to meet every performance requirement.

Two impellers can be manufactured as one for parallel operation, two single stage impellers back-to-back, a double suction impeller as shown in Figure 4.35. A double suction pump should have a lower NPSHr than an equivalent single suction pump. For liquids contaminated with solids or sewage and where production must be simplified due to manufacturing problems of the tough materials used, the design principle of "greatest possible efficiency" is discarded and the impeller is given a much simpler shape. A distinction is made in theory between closed, semi-open and open impellers as well as special shapes for contaminated liquids. See also "non-clogging" pumps.

4.2.3.2 Multi-stage pumps

The efficiency of pumps is dependent upon the specific speed, as shown in Figure 4.36. Single stage pumps, required to produce high differential heads or handle low flows will have low specific speeds when calculated by equation 4.15. High differential heads can be produced by using several low head impellers in series. The specific speed will now be higher because the flow is the same as before but the differential head per im-

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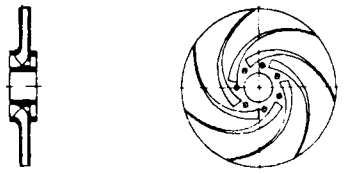
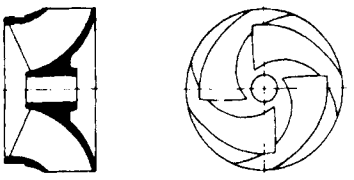
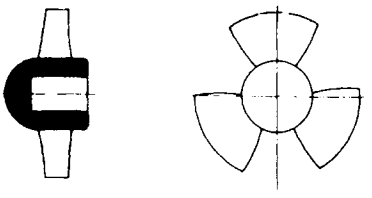


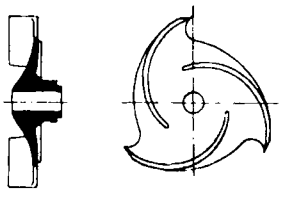
Closed centrifugal impeller. The illustration on the right shows the impeller with the front shroud removed



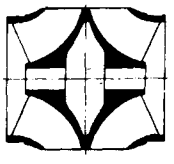
Closed mixed flow impeller. The illustration on the right shows the impeller with the front shroud removed



Axial impeller (propeller)



Open centrifugal impeller



Double suction mixed flow impeller
Figure 4.35 Examples of different impeller types

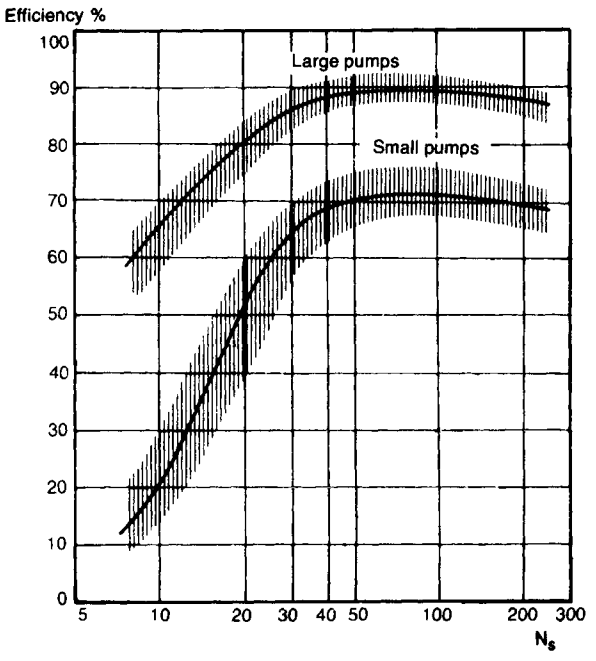


Figure 4.36 Pump efficiency as a function of specific speed

peller is smaller. A multi-stage pump consists of several similar impellers working on the same liquid consecutively. Doubling the number of impellers increases the specific speed by $2^{0.75}$ = 1.68. The following efficiencies, for example, can be read off from Figure 4.36 for a small pump:

No of stages	N_s	Efficiency %
1	10.0	21
2	16.8	42
4	28.2	63

The range of application of multi-stage pumps for delivery heads over 50 and up to 100 m, is, as a general rule, dependent upon the flow and speed.

4.2.4 Classification of rotodynamic pumps by application

In addition to a classification in accordance with different hydraulic designs, it is practical to apply a classification for different areas of application. Within each such area, there are special design and construction features dependent upon the:

- Liquid properties
- Allowable leakage
- Driver type
- Installation arrangement
- Allowable noise level
- Operational safety

The rotodynamic pump Sections 1.3 and 1.4 of Chapter 1 should be studied individually to assist in the selection of the appropriate pump. These pump types have been classified according to detailed technical and marketing considerations.

4.3 Positive displacement pumps

4.3.1 Rotary positive displacement pump theory

Basic principles

One of the simplest rotary pumps, and probably the easiest to understand, is the external gear pump, see Figure 4.37. One gear is driven by an external power source, the other meshes with the driven gear. The gears are enclosed in a close fitting casing. The radial clearance around the gear teeth is controlled by machining; the axial clearance down the sides of the gears is usually controlled by shimming during assembly. Clearance is necessary to avoid seizure during operation. The axial clearance is also necessary to allow lubrication.

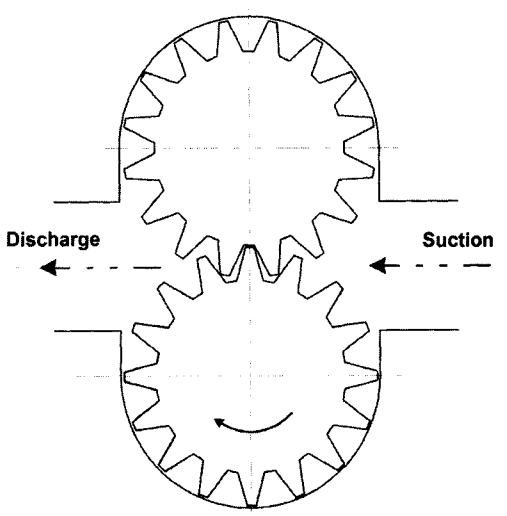


Figure 4.37 Diagrammatic external gear pump

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Liquid is carried around the periphery of the casing in the voids between the teeth. The theoretical flow is given by:

$$Q_{th} = 0.001 \times a \times w \times (2 \times N) \times rpm/60 \qquad \text{Equ 4.24}$$

where:

Q_{th}	=	theoretical flow (cc/s)
a	=	area between teeth (mm^2)
w	=	tooth width (mm)
N	=	number of teeth in each gear
rpm	=	speed (1/minute)

Equation 4.24 applies to gear pumps where identical gears are used; this is not essential for gear pump designs but it is the most usual. Theoretical flow cannot be realised in practice. Leakage occurs directly from discharge to suction where the gears mesh; this leakage can be very small and is dependant upon the tooth form and the clearance. Most leakage occurs around the periphery between the tips of the teeth and the casing. This leakage is complex to analyse but approximates as follows:

$\propto$	differential pressure
$\propto$	gear width
$\sim\propto$	tip clearance^3
$\propto$	$1/\mu$
$\propto$	1/N
$\propto$	1/gear tip length
$\sim\propto$	1/function of tip speed

Gear pumps rely on viscosity to reduce leakage. As viscosity reduces the leakage becomes a greater proportion of the theoretical flow. The gear pump characteristic shown in Figure 4.48 in Section 4.3.3 indicates the effect of speed, viscosity and differential pressure. Leakage is reduced at higher speeds for constant viscosity and differential pressure.

A popular light oil grade for centrifugal pumps with rolling bearings is ISO 32. At 20 °C these oils would have a viscosity of 85 mm^2/s. Water has a viscosity about 1 mm^2/s. It can easily be seen in Figure 4.48 how the leakage losses escalate as less viscous liquids, more like water, are pumped.

The mechanical efficiency of gear pumps is controlled by several mechanisms and assumptions should not be made when calculating power required. At low viscosities the efficiency tends to improve with increasing speed. At high viscosities the efficiency may be constant or reduce with increasing speed. The distinction between high and low viscosity is a function of pump size. Mechanical efficiency usually increases as the differential pressure increases. Small pumps can have efficiencies of 25% and large pumps 70%.

Because gear pumps have multiple pumping elements, for example, the teeth, the flow pattern may have a slight ripple which can cause pressure pulsations. Pumps with few teeth are worst. Tests by Professor Edge, of Bath University, have indicated peak-to-peak pressure pulsation levels of 3.25 bar with a mean suction pressure of 3 bara.

Wear in rotary displacement pumps

The internal leakage varies as the cube of the clearance between the rotating components and the casing. Displacement pumps are thus susceptible to increased clearances due to wear. Displacement pumps used for "abrasive" liquids are constructed from wear-resistant materials and operated at lower speeds. Exceptions to these guidelines can be made by utilising principles of construction which eliminate leakage by the use of an elastic element, peristaltic pumps, for example.

Considering elementary principles, wear is a function of:

- Surface pressure between components
- Relative rubbing speeds between components
- A dependance upon the coefficient of friction

or with more usual wear theory, the combination of surface pressure and rubbing speed, PV.

The heat produced per unit of surface area is dependant upon PV and the coefficient of friction and can be a significant factor.

The technique of creating wear-resistant rotary pumps is concerned with reducing the product PV. Low values of velocity can result in large pumps, the only solution then is to reduce the surface pressure.

The driving force for rotary pumps can be transmitted to the various components in different ways. In the case of gear pumps the external drive is normally applied to only one of the gears, while the other is driven by engagement with the gear teeth of the first. However, if both the gears are driven externally, the load relationship within the process liquid is completely different. It is necessary to synchronise the shafts by means of a separate synchronising gear so that the teeth pass within each other when rotating without contact. The surface pressure between the interactive gear components is thus reduced to nil. The shape of the interactive gear surfaces can now be chosen with regard to the pumped fluid. Instead of gear teeth, the pump elements can now be given smoother shapes or radically different shapes, as in the case of lobe pumps and rotary piston pumps, (see Chapter 1, Section 1.5.8).

Within the rotary pump group there is a special "family" of pumps whose interactive components are guided by external devices as in the case of the lobe pump mentioned above. Even pumps where the rubbing pressure is eliminated in other ways also belong to this group. Examples of such pumps are rotary piston pumps and twin screw pumps. (See Chapter 1, Section 1.5.4.)

With the rubbing pressure eliminated or greatly reduced the duty range can be increased. Depending upon other design features, this range of application can be extended to contaminated liquids, low viscosity, some dry-running capability or high speed.

Rotary pumps suitable for abrasive liquids should be constructed according to the following:

- The active components should be positively located by external means or have greatly reduced sliding pressure.
- The forces generated on the pumping elements by the liquid pressure should be absorbed by bearings which are not in contact with the liquid.

4.3.2 Reciprocating positive displacement pump theory

The theoretical principles of positive displacement pumps are basically very simple. For each working cycle, each stroke, a certain volume of liquid is drawn into and expelled from the cylinder virtually irrespective of the differential pressure. The volume of liquid depends upon on the displacement (swept volume) and the volume of the cylinder. The maximum attainable pressure is primarily dependent only upon the mechanical construction and the driving power available. The maximum pressure can be limited by means of a pressure relief valve installed in the system or integrated in the pump. Because reciprocating pumps can develop such high pressures they must always be operated with over-pressure protection. A relief valve or bursting disc must be fitted. In hydraulic power applications it has become standard practice to incorporate the over-pressure protection with other functions, such as unloading, by using a common pilot-operated valve. In process applications this approach would not be tolerated; a dedicated device must be fitted solely for protection.

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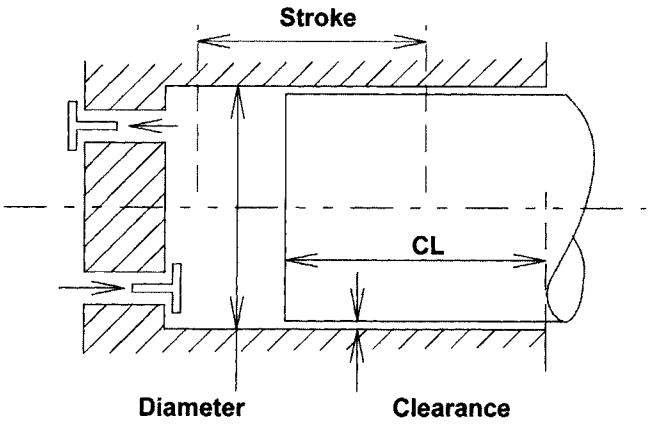

Figure 4.38 Basic reciprocating pump

Consider the single cylinder pump shown in Figure 4.38. Ignoring leakage and compressibility, the ideal flow delivered is calculated from:

$$Q_{th} = 0.001 \times \text{displacement} \times n/60$$

and

$$\text{displacement} = s \times \pi d^2/4 \qquad \text{Equ 4.25}$$

where:

Q_{th} = theoretical flow (cc/s)

n = strokes per minute (1/minute)

d = piston diameter (mm)

s = stroke (mm)

The displacement of the piston establishes the name of the pump type, positive displacement. Displacement is also called swept volume as it is the volume swept out by the face of the piston.

The ideal power required for the pump is given by:

$$P_{th} = Q_{th} \times (p_d - p_s) \qquad \text{Equ 4.26}$$

where:

P_{th} = ideal power (W)

Q_{th} = ideal flow (m^3/s)

p_d = discharge pressure (gauge) (Pa)

p_s = suction pressure (gauge) (Pa)

Equation 4.26 is not very useful for engineering applications. The Pascal is an extremely small unit of pressure, the kPa is very small. In order to avoid confusion at site, where suction pressure gauges may be calibrated in kPa and discharge gauges in MPa, the pump industry prefers to use the "bar". 1bar = 100kPa. The following equations are more practical.

$$kW = \frac{m^3/h \times \Delta bar}{36} \qquad \text{Equ 4.27}$$

$$kW = \frac{L/s \times \Delta bar}{10} \qquad \text{Equ 4.28}$$

The ideal power is based on ideal flow. This Section on theory commenced by excluding leakage and compressibility. Obviously real pumps leak and real liquids are compressible. Compressibility is the easiest to tackle.

It can be seen from Figure 4.38 that the piston stroke does not extend right to the end of the cylinder. The volume of liquid trapped in the cylinder, together with liquid compressibility, affects the volumetric efficiency. The volumetric efficiency is the ratio of the liquid volume pumped to the displacement of the piston. We have already seen that the discharge volume may be smaller than the suction volume, this leads to two definitions of volumetric efficiency — suction and discharge.

$$ve_S = 100 \times \left(\frac{\left(1 - \Delta p \times \chi \times \left(1 + \frac{V_{int}}{V_{sw}}\right)\right)}{(1 - \Delta p \times \chi)} \right) \qquad \text{Equ 4.29}$$

$$ve_D = 100 \times \left(1 - \Delta p \times \chi \times \left(1 - \frac{V_{int}}{V_{sw}}\right)\right) \qquad \text{Equ 4.30}$$

$$ve_S = 100 \times \left(1 - \frac{V_{int}}{V_{sw}} \times \left(1 - \frac{\rho_D}{\rho_S}\right)\right) \qquad \text{Equ 4.31}$$

$$ve_D = 100 \times \left(1 - \left(\left(\frac{V_{int} + V_{sw}}{V_{sw}}\right) \times \left(1 - \frac{\rho_S}{\rho_D}\right)\right)\right) \qquad \text{Equ 4.32}$$

where:

ve_S = suction volumetric efficiency (percentage)

ve_D = discharge volumetric efficiency (percentage)

Δp = differential pressure (bar)

χ = compressibility (*)

V_{int} = clearance volume (cc)

V_{sw} = swept volume (cc)

ρ_D = discharge density of liquid (kg/m^3)

ρ_S = suction density of liquid (kg/m^3)

(*) the units for compressibility are "contraction per unit volume/bar differential pressure".

Compressibility is the inverse of the bulk modulus which has units of N/m^2. The bulk modulus of water at 20 °C from atmospheric pressure to 100 barg is approximately 2.1 x 10^4 bar. The presence of entrained and dissolved gas or air in the liquid increases the compressibility and reduces the volumetric efficiency.

The formulae for volumetric efficiency indicate the importance of keeping the clearance volume as small as possible. Also the volumetric efficiency reduces as compressibility increases. At low differential pressures the compressibility of water can be used for most other liquids without creating significant errors. Some liquids are much more compressible than water, liquid gases for example can be six times as compressible, and using water compressibility will create problems of undersizing and low flow.

Leakage further reduces the actual liquid volume delivered by the pump. The term "slip" is used to collect together all the liquid losses which can be evaluated and those which must be compensated by an experimental allowance. Slip is evaluated as a percentage and subtracted from the volumetric efficiency.

Because there is a clearance around the periphery of the piston, leakage can occur due to the pressure rise within the cylinder; this leakage can be approximated from:

$$Q_L = 0.00156 \times \left(\frac{d}{CL}\right) \times \frac{h^3 \times \Delta p}{\mu} \qquad \text{Equ 4.33}$$

where:

Q_L = leakage flow (cc/min)

d = cylinder bore (mm)

CL = contact length (mm)

h = clearance (mm)

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Δp = differential pressure (bar)

μ = viscosity (cP)

As the piston and cylinder wear the leakage rate rises dramatically. The liquid viscosity may be lower in the clearance due to frictional heating. This leakage is fairly constant and independent of pump speed, strokes per minute.

The valves shown diagrammatically are spring-loaded automatic valves. The differential pressure across the valve, in the direction of flow, opens the valve and liquid flow forces hold the valve open. As the flow reduces the valve starts to close due to the spring force. The pressure losses and leakage through valves of this generic type are impossible to generalise due to the infinite range of proportions and shapes. However, the following relationships can be used for guidance:

- For fixed valve lift

  $\Delta p \propto Q$

  $\propto \mu$

- For fixed flow

  $\Delta p \propto m$

  $\propto l/\text{lift}^3$

One valve loss, particularly troublesome at high speed, is valve backflow. Automatic valves are closed by the spring; the spring applies a force to the mass of the valve and the valve accelerates. At higher speeds, there is insufficient time for the valve to close, unless a stronger spring is fitted. The valve remains slightly open when the piston travel has reversed allowing some liquid to escape. After inception, this loss varies with speed and is almost linear.

Some pumps are fitted with actuated valves, or valve timing is incorporated in the rotating cylinder block of hydraulic pumps. These valves are usually rotary and driven directly by the main shaft. The valve timing is optimised for the design conditions. Operating at other conditions causes backflow or over-compression.

Valve flow losses are complicated because the liquid flow is approximately sinusoidal. The reciprocating pumps described here are usually used for hydraulic power applications. The liquid is oil or water-oil emulsion providing good lubrication for the close fitting piston. The pumps are axial or radial piston. The pistons are driven by a mechanism which produces simple harmonic motion. The piston dynamics, sine and cosine functions, are shown in Figure 4.39.

The liquid is unable to follow the piston dynamics exactly due to inertia. The liquid inertia causes the liquid to accelerate and decelerate faster than the piston, causing pressure pulsations. Liquid flow variations and pressure pulsations caused by reciprocating pumps can not be approximated easily as they are a function of the system as well as the pump. In critical applications, and with low viscosity liquids, an acoustic analysis should be performed which will evaluate pressure pulsations, flow variations and acoustic resonance.

Two variations of reciprocating pump design used for process applications where the liquid may not have good lubricating properties, are shown diagrammatically in Figure 4.40. The plunger pump is single acting whereas the piston pump is double acting.

The reciprocating motion is produced by a crankshaft with connecting rods and crossheads, the slider-crank mechanism. There is no side thrust applied to the piston or plunger from the driving mechanism, this is applied by the crosshead to the crankcase. The length of the connecting rod compared to the throw of the crankshaft is important because it affects the piston/plunger dynamics and the crosshead side thrust. The sine and cosine functions of simple harmonic motion are modified by connecting rod proportions as shown.

$$x = r(1 - \text{Cos}\theta) + l\left(1 - \sqrt{1 - \lambda^2 \text{Sin}^2\theta}\right) \qquad \text{Equ 4.34}$$

$$v = \omega r\left(\text{Sin}\theta + \frac{\lambda\ \text{Sin}\theta\ \text{Cos}\theta}{\sqrt{1 - \lambda^2 \text{Sin}^2\theta}}\right) \qquad \text{Equ 4.35}$$

$$a = \omega^2 r\left(\text{Cos}\theta + \frac{\lambda\left(1 - \text{Sin}^2\theta\left(2 - \lambda^2 \text{Sin}^2\theta\right)\right)}{\left(1 - \lambda^2 \text{Sin}^2\theta\right)^{3/2}}\right) \qquad \text{Equ 4.36}$$

where:

x = travel (m)

v = velocity (m/s)

a = acceleration (m/s/s)

r = crank throw (m)

l = connecting rod centres (m)

ω = angular velocity (rad/s)

λ = r/l (non-dimesional)

θ = crank angle (degree)

The three parameters are plotted in Figure 4.41, for λ=0.25, alongside the simple harmonic motion curves. Notice that the:

- Curve shapes are modified
- Maximum velocity is not at 90 °

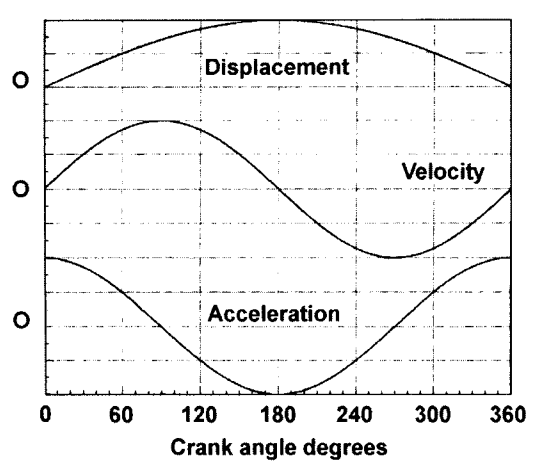

Figure 4.39 Simple harmonic motion piston dynamics

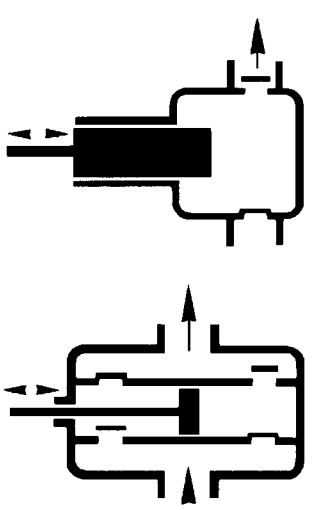

Figure 4.40 Process plunger and piston pumps

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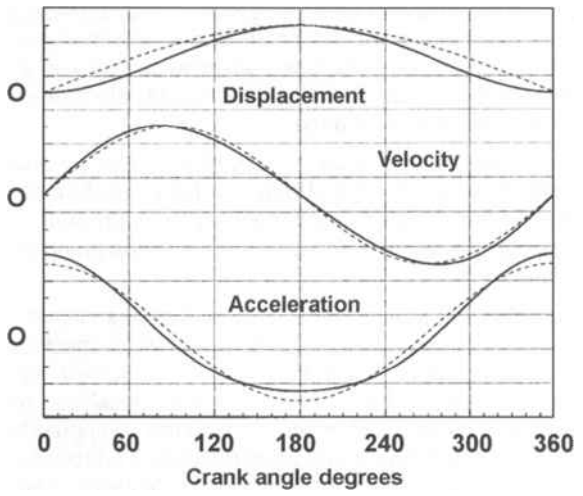

Figure 4.41 Travel, velocity and acceleration of slider-crank mechanism

- Forward and return acceleration is different
- Maximum acceleration is greater

Having an infinitely long connecting rod produces simple harmonic motion. In practice, connecting rods are 4 to 7 times the crankshaft throw.

A plunger pump is very similar to the piston pumps described previously for hydraulic fluid power except a stationary sealing stuffing box prevents most leakage. The piston pump shown is double acting and has moving seals on the piston as well as a stationary sealing stuffing box. The prime distinction between piston and plunger pumps is the absence of moving seals in the plunger pump. Plunger pumps are not a type of piston pump.

Piston pumps are used for pressures up to about 150 barg and plunger pumps for higher pressures. There is no absolute limit on plunger pumps, 10000 barg is possible, but popular pumps operate up to 1200 barg. In Chapter 1, Figure 1.90 shows a piston pump for about 25 barg and Figure 1.91 shows a plunger pump for about 350 barg . Figure 1.92 shows a plunger pump built as vertical pump.

The stuffing box prevents leakage of process liquid to the atmosphere. Simple seal designs are used for non-hazardous liquids such as water and ethylene glycol, Figure 4.42. Hazardous liquids; petrol, methanol, LPG; or liquids contaminated with hazardous gases like hydrogen sulphide, can use very complicated stuffing boxes depending upon the location, Figure 4.43.

Auxiliary systems can be fitted to stuffing boxes, see Section 8.5, in Chapter 8.

Reciprocating pumps for process applications can be fitted with various valve designs to suit the liquid handled, Figure 1.96, Chapter 1.

Mechanical losses

Reciprocating pump crankcases and actuating mechanisms are designed to exert a specific force on the piston or plunger, the rod load. Pumps are selected so that they operate as close to the design rod load to minimise mechanical losses. Different piston/plunger diameters are fitted to match the operating pressure. Figure 4.44 indicates the effect of discharge pressure on a 3" stroke plunger pump, with a fixed plunger diameter and a simple single packed stuffing box as shown in Figure 4.42.

At pressures below the design pressure, friction becomes a more significant influence and reduces the mechanical efficiency. In general, the mechanical efficiency of reciprocating pumps is not affected by pump speed or viscosity. Larger pumps may be more efficient; pumps of 1000 kW can have a mechanical efficiency of 94% at design pressure. Complicated,

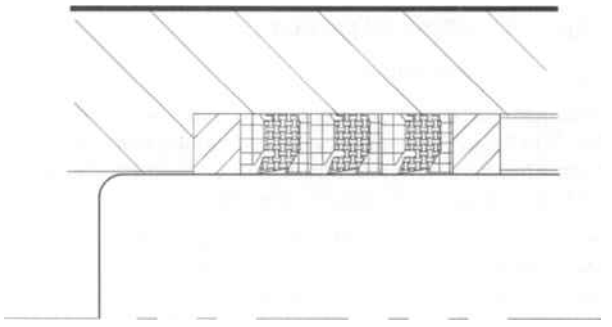

Figure 4.42 Non-adjustable stuffing box for water

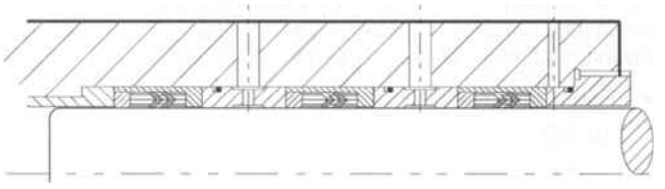

Figure 4.43 Self-adjusting stuffing box for hazardous liquids

multi-seal stuffing boxes increase friction and reduce the mechanical efficiency.

Wear in reciprocating pumps

The main wearing parts in reciprocating pumps are piston seals, stuffing box packing and valves. Wear in stuffing box packing produces an external leak which operators or maintenance staff can recognise and act upon. Wear in piston packing and valves produces internal leaks, increased slip, and can only be detected as a reduction in pump flow.

Reciprocating pumps can handle liquids contaminated with solids. The Miller Number should be evaluated. Piston pumps in standard materials are adequate for Miller Numbers up to 50. Above 50, severe wear problems may occur. Plunger pumps can handle the most abrasive liquid/solid combinations by running at reduced speeds, using ball valves and flushing the stuffing box to protect the packing.

Both piston and plunger pumps can be modified for solids handling by introducing a diaphragm to completely isolate the reciprocating driving element from the process liquid. Diaphragm pumps without pistons or plungers are also possible. Diaphragms can be driven directly by another fluid, liquid or gas, which has timed control valves to regulate the motion.

Flow and pressure pulsation

For many positive displacement pumps the flow varies during the period of the working cycle. Extreme examples of flow variation occur in single and twin cylinder reciprocating pumps, see Figure 4.45.

During a working cycle the instantaneous flow varies, to a greater or lesser extent, for most positive displacement pump types. In order to reduce variations in flow it is possible to increase the number of cylinders or elements or to provide pulsation dampers at the suction and discharge connections. These can be filled with air/gas or equipped with air cushions or springs separated from the liquid by an elastic membrane. The size of the flow variation is an important feature, which unfortunately cannot be easily specified by the manufacturer. The flow variation, and the resulting pressure pulsations, are a function of the pump design, the liquid properties and the installation design. The same pump will have different characteristics in a different installation.

Flow variations and pressure pulsations measured during factory testing will not be representative of values experienced at site, unless the pipework and liquid conditions are identical. Flow resistance in a test rig consists largely of throttling losses in control valves i.e. with a resistance varying as the square of the flow. Dampening in rubber hoses for example, also has

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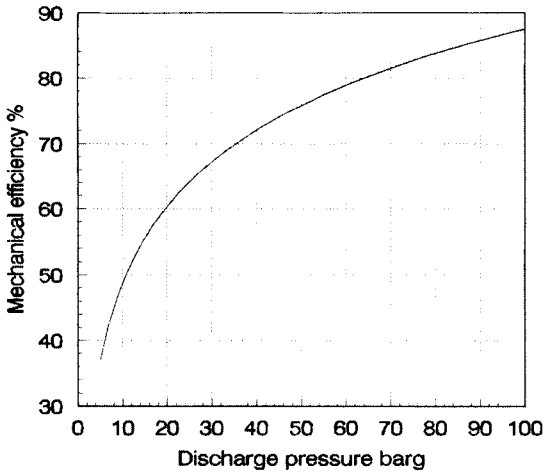

Figure 4.44 Discharge pressure and mechanical efficiency

considerable influence. If flow variations and pressure pulsations are critically important for an installation, then a computer simulation, analogue or digital, must be performed of the complete system to verify instantaneous flow and pressure values together with sources of acoustic resonance.

NOTE: The flow variations shown in Figure 4.45 are based on an incompressible liquid, an ideal pump and no suction or discharge systems. These conditions cannot be reproduced in practice. Flow diagrams of this type are purely indicative and very approximate. No practical data can be derived from such diagrams and no useful calculations can be performed for flow variations, pressure pulsations or pulsation damper sizing.

The minimum flow for single and twin cylinder pumps is zero so the total flow variation is 100%. Screw pumps have flow variations of a few percent. Pressure pulsations caused by flow variations are the primary cause of noise and vibration in positive displacement pump piping. In the case of gear pumps for example, it is very easy for a small volume of liquid to become trapped in a pocket at the moment of gear contact, the volume of which changes during the course of rotation. This gives rise to harsh running and high noise levels. The problem can be considerably reduced by introducing a slight helix angle on the gear; by providing relief grooves in the side plates or by increasing the side clearance.

4.3.3 Positive displacement pump curves

Figure 4.46 shows a positive displacement pump curve drawn in the same manner as a rotodynamic pump curve. Little detailed information can be gained from the curve and two curves must be drawn separately to show the power relationship.

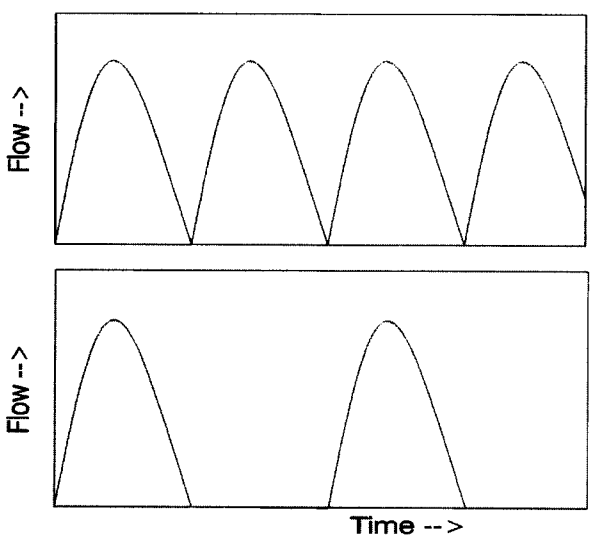

Figure 4.45 Flow variation of single and twin cylinder single acting reciprocating pumps

If curves are drawn it is better to plot the discharge pressure along the abscissa. This then conforms to mathematical standards; the abscissa being the independent variable. Flow, power and mechanical efficiency can be plotted on the same curve, see Figure 4.47. For sensible flow information to be retrieved an expanded scale must be used and zero flow is not necessarily shown. The characteristic curve of flow v pressure may not be linear. Compressibility and slip effects may be more pronounced at higher pressures. Curves for different speeds may not be the same shape. When various liquids are pumped, or operating conditions vary, additional curves may be required. For fixed speed pumps with a single duty point, tabulated data is preferred.

Figure 4.48 shows the change in fixed speed pump flow resulting from changes in differential pressure and liquid viscosity for an external gear pump. It can be seen that the reduction in flow is very small for relatively large changes in differential pressure and is affected by the liquid viscosity.

As mentioned earlier, viscosity and speed can play an important role in the performance of rotary pumps such as gear pumps. Figure 4.49 indicates the speed, viscosity and differential pressure effects on a gear pump.

To make pump selection easier performance curves can be drawn differently.

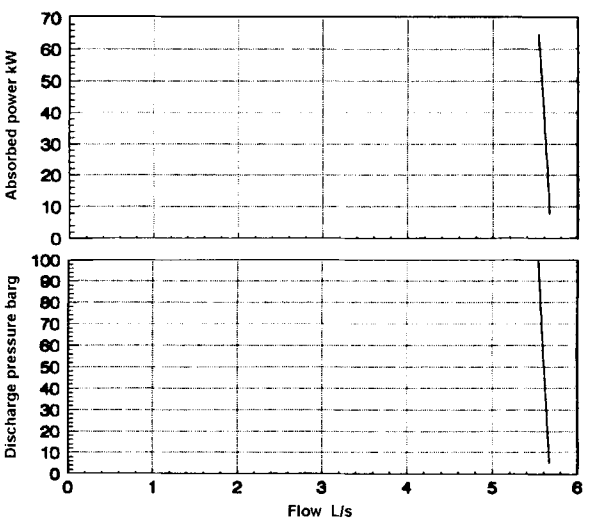

Figure 4.47 Performance curve for positive displacement pump

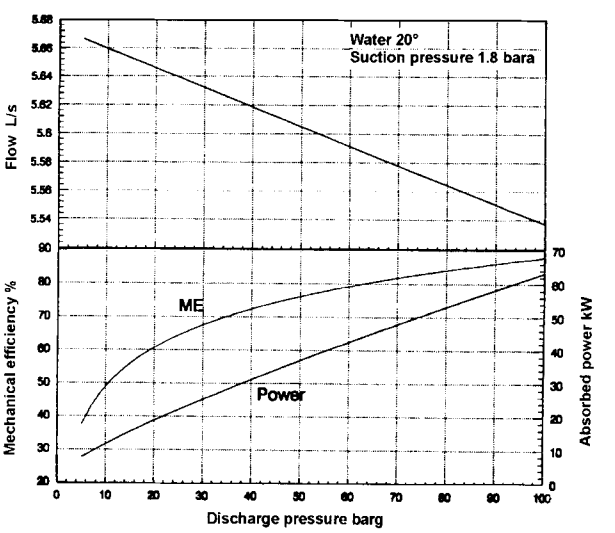

Figure 4.47 Performance curve for positive displacement pump

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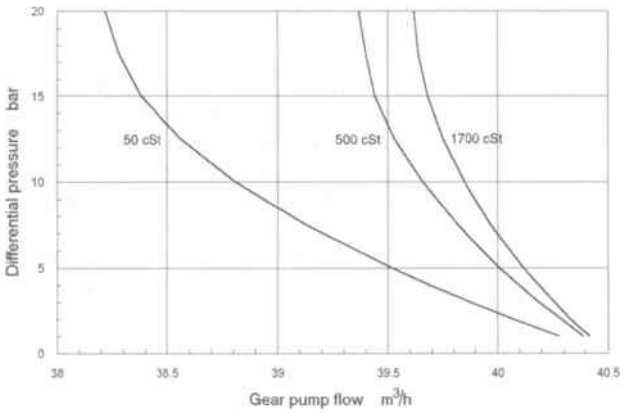

Figure 4.48 The effects of slip on gear pump flow

Figures 4.50 and 4.51show the curves for a lobe pump for two different viscosities.

4.3.4 Types of positive displacement pumps

Rotary positive displacement pumps are generally self-priming i.e. they can begin pumping with a "dry" suction pipeline and pump casing. Reciprocating positive displacement pumps are self-priming, in theory, although they tend to be poor compressors. In general, reciprocating pumps should be vented and primed prior to operation unless the manufacturer specifically states that self-priming is allowable.

All positive displacement pumps can pump a wide variety of liquids of varying characteristics, from very low viscosity liquids to very viscous pastes. The construction, application and pumping characteristics of different types of displacement pumps varies considerably however. It is therefore necessary to understand thoroughly the specific characteristics of each type of pump in order to be able to determine the suitability of a particular pump for a particular application. Gear pumps are not suitable for abrasive liquids for example. A progressive cavity pump with its flexible stator is more suitable. If the discharge pressure is relatively high a vane pump with flexible vanes is unsuitable, a gear pump or piston pump being a better choice.

The specified volume flow per working cycle is relatively constant and reduces insignificantly for moderate pressure increases. Liquids with high compressibilities, liquid gases, should always be evaluated for density changes. Control of the flow should be carried out by increasing or decreasing the pump speed. The flow can also be controlled by arranging a by-pass between the discharge and suction source. Do not, however, attempt to throttle the discharge in order to reduce the flow, as in the case of centrifugal pumps. This results in increased pressure, increased power consumption and increased loads on the working components, without significantly reducing the flow.

Serious damage and injury to personnel can occur if the pump discharge should become completely blocked, by the closing of a valve for example. The pressure will increase rapidly and can become so great that it causes failure of pipelines and other vital components. A dedicated pressure relief device should always be fitted as a safety precaution. The opening pressure of which should be selected so as to provide a good margin of safety for the protection of the system and its components.

A certain amount of internal leakage (slip) is always present in displacement pumps, this is due to the clearance between the working elements and inefficiencies of internal valves. The clearance can be of the order of 0.01 to 0.9 mm and is necessary in order to counteract galling and to reduce friction within the pump. Leakage occurs from the discharge side back to the suction side, i.e. against the direction of flow. The amount of slip, which reduces volumetric efficiency, varies greatly according to the type of pump construction. A progressive cavity pump

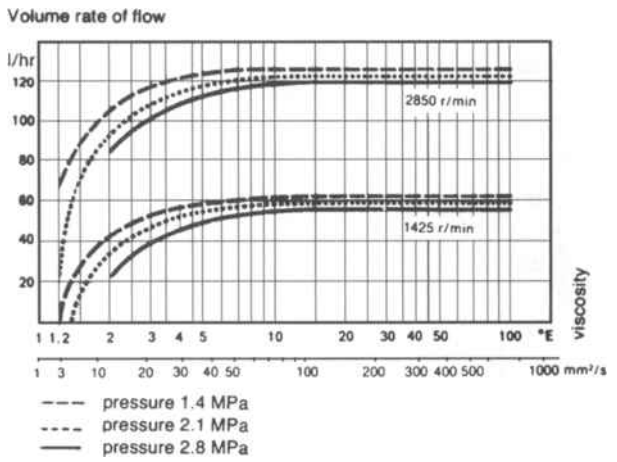

Figure 4.49 The effect of viscosity, speed and dp on a small gear pump

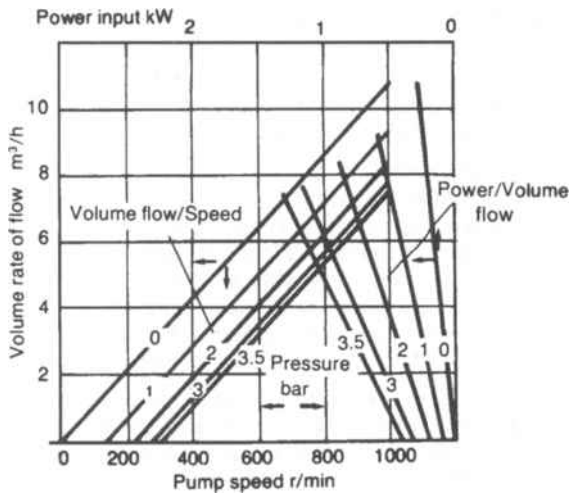

Figure 4.50 Performance curve for a lobe pump in water

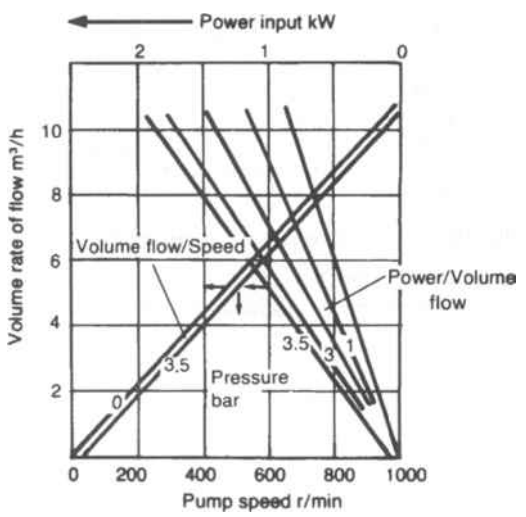

Figure 4.51 Performance curve for the lobe pump in Figure 1.70, Chapter 1, on liquid at 100 cSt

with its long rotor and effective sealing rubber stator has less slip than a conventional gear pump having only line contact between the gears. The pumped liquid, if it is sufficiently viscous, can help to "seal" the clearances and reduce internal leakage.

Positive displacement pumps should not be installed to make use of their maximum suction lift capacity. Pump wear is unavoidable and this means that in time, the maximum suction performance will be reduced. This means that the pump loses its priming ability with subsequent loss of its pumping function. Unnecessary service costs are thus incurred in order to constantly maintain the pump in an "as new" condition.

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4.3.5 Classification of positive displacement pumps by application

As discussed in Section 4.3.4, there are many types of positive displacement pumps. The rotodynamic pump, Sections 1.3 and 1.4 of Chapter 1, should be studied individually to assist in the selection of the appropriate pump. These pump types have been classified in accordance with their mode of operation.

NOTE: Reciprocating pumps for hydraulic power applications usually have no seals and rely on the pumped liquid for all lubrication. Reciprocating pumps for process applications do not rely on the pumped liquid for any lubrication. Process pumps can be used for hydraulic power applications.

4.3.6 Classification of other pump types by application

As mentioned in Sections 4.2.4 and 4.3.5, the 'other pump' types in Section 1.6 of Chapter 1, should be studied individually to assist in the selection of the appropriate pump. These pump types have also been classified in accordance with their mode of operation.

4.4 Suction performance

4.4.1 Cavitation

The word cavitation originates from the Latin meaning "hollow". Cavitation occurs when the static pressure somewhere within the pump falls to or below the vapour pressure of the liquid. Some of the liquid then vaporises giving rise to the formation of vapour bubbles. These vapour bubbles are entrained in the liquid and transported until they reach an area where the static pressure is greater than the vapour pressure.

At this point the vapour bubbles implode since they can no longer exist as vapour. Each implosion causes a severe pressure wave. The continuous repetition of this sequence of events occurring many times and at high frequency can cause mechanical damage to the pump materials. Furthermore, the hydraulic performance of the pump is reduced by the onset of cavitation. Cavitation is therefore an undesirable phenomenon and should be avoided.

The severity of cavitation damage is related to the thermodynamic properties of the liquid and the pump operating conditions. The important factors to consider are:

- Latent heat of vaporisation
- Differential pressure
- Rate of pressure rise
- Volume of liquid in component
- Volume of vapour
- Material of component
- Time of exposure

Water tends to be the worst liquid for cavitation damage because the latent heat of vaporisation is large compared to other liquids. The time of exposure is very important. If a pump cavitates for a few seconds during start-up the reliability and performance may not suffer noticeably. If a pump operates with cavitation for hours the likelihood of damage is increased. Figure 4.52 shows mild cavitation damage on a centrifugal pump impeller vane. Figure 4.53 shows severe cavitation damage to a centrifugal pump impeller.

Figure 4.54 shows cavitation damage on a reciprocating pump valve. The damage is very unusual because it is on the outside of the seat taper. The valve was probably dislodged by the forces generated by the cavitation and then the damage occurred due to the high velocity liquid flowing around the seat.

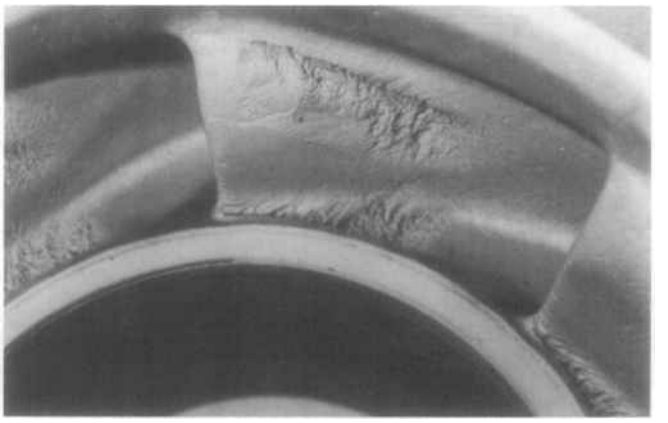

Figure 4.52 Mild cavitation damage on a centrifugal pump impeller vane

Figure 4.53 Severe cavitation damage to a centrifugal pump impeller

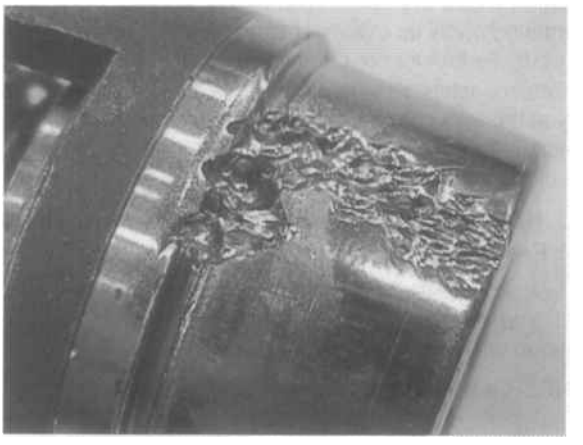

Figure 4.54 Cavitation damage on a reciprocating pump valve

The risk of cavitation is greatest where the pressure within the pump is lowest. The lowest pressure in a centrifugal pump occurs on the suction side of the impeller blades, slightly downstream of the front edges where the liquid has started to accelerate, see Figure 4.55. At the pump's inlet connection at the same elevation as the shaft the pressure is somewhat greater, by Δp. At this point, which lies at the static suction head h_s above the liquid free surface, the flow velocity is c_s. By applying Bernoulli's equation, see equations 3.4 and 3.5, in Chapter 3, the flow through the suction pipe is:

$$p_s = p_a - \rho h_s - \rho g h_{fs} - \rho \frac{c_s^2}{2} \qquad \text{Equ 4.37}$$

$$p_{min} = p_s - \Delta p \qquad \text{Equ 4.38}$$

where:

p_{min} = lowest pressure within the pump (N/m^2)

p_s = suction pressure (N/m^2)

Δp = local pressure reduction (N/m^2)

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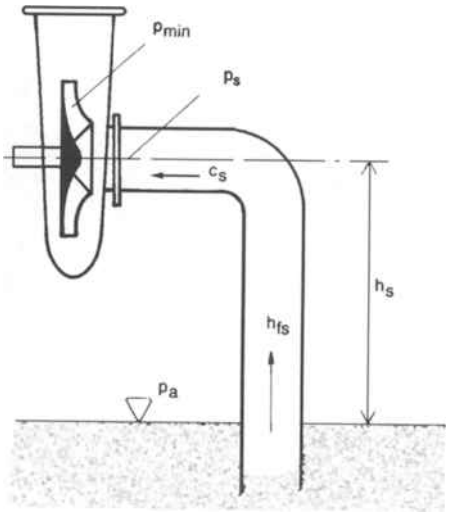

Figure 4.55 Cavitation model

c_s	=	Flow velocity in inlet pipe (m/s)
h_s	=	Static suction head (m)
h_{fs}	=	Friction losses in suction pipe (m)
p_a	=	Pressure on liquid surface (N/m^2)

Δp covers any losses within the pump, after the suction connection, such as velocity increases or friction losses.

Positive displacement pumps require a corresponding pressure differential, Δp above vapour pressure, in order to avoid cavitation within the pump. The lowest pressure in a pump is determined partly by external factors such as suction pressure and partly by factors concerned with construction and design. The latter consists of the local pressure drop Δp and the flow velocity at the inlet connection c_s. In order to avoid cavitation it is necessary to ensure that p_{min} is greater than the vapour pressure of the liquid p_v.

4.4.2 Net Positive Suction Head and Net Positive Inlet Pressure

The concept of NPSH, Net Positive Suction Head, is encountered in connection with cavitation in rotodynamic pumps . The definition of NPSH is as follows:

NPSH = suction pressure - vapour pressure

$$NSPH = \frac{p_s - p_v}{\rho g} \qquad \text{Equ 4.39}$$

In this context the suction pressure is the pressure at the NPSH datum. For rotodynamic pumps the NPSH datum is the shaft centre-line, except for vertical pumps when it is the front face of the impeller, see Figure 4.56.

In some cases when purchasing a new pump, the user will not know where the NPSH datum is going to be. In these cases, the NPSHa and the suction pressure should be referred to the top of the foundations. NPSH is a differential head, therefore the pressures are converted to metres of liquid when required.

Some textbooks state that NPSH refers to the suction total pressure, the suction stagnation pressure. This makes life very complicated. The pump suction pressure is the static pressure; head has been deducted to produce the velocity head of the flowing liquid. This is the value to which the pump differential head is added. It does not make sense to calculate a different suction pressure, the stagnation pressure, for NPSH purposes.

NPSH has two forms:

- The system form
- The pump form

The system form is designated NPSHa, Net Positive Suction Head Available. NPSHa is calculated from the suction system configuration and is the suction pressure minus the vapour pressure at the NPSH datum. NPSHa tells the pump manufacturer how much Δp is available for losses in the pump suction.

The pump form is designated NPSHr, Net Positive Suction Head Required. NPSHr is a characteristic of the pump and specifies how much Δp is necessary for losses in the pump suction.

NPSHa and NPSHr are the difference between two pressures or heads; there is no requirement to specify gauge or absolute.

Both forms of NPSH are variables, not constants. Both are a function of flow. In a fixed suction system configuration, NPSHa will increase as flow is reduced because friction losses and the velocity head requirement are reduced. Conversely, NPSHa will reduce as flow increases because the same losses increase. NPSHr for a pump is predicted during the design phase and confirmed by testing. NPSHr increases with flow and, at some point, increases as flow is reduced.

Rotodynamic pumps are optimised for a specific operating condition; flow and differential head; the Best Efficiency Point (BEP). When operating at other conditions parasitic losses increase and blade/liquid angles are mismatched. As flow is reduced below BEP angular mismatch increases. At a certain point, the losses created by mismatch become larger than the gains produced by reduced friction losses and NPSHr increases, see Figures 4.12 and 4.13.

In an ideal world, the NPSHr value for a pump would signify the initiation of cavitation. However it is extremely difficult, if not impossible, to detect the first indications of cavitation. Because of this problem various criteria have evolved as bench-marks for defining cavitation. The most popular definition is for 3% head loss. The pump is tested at constant flow and the suction pressure is reduced until the differential head has decayed by 3%, see Figure 4.57. The NPSHr is calculated from the suction pressure and the vapour pressure.

A complication to the 3% head loss criterion arises with multi-stage pumps. 3% head loss of what ? Some manufacturers use 3% of the pump differential head. In fact it should be 3% of the first stage differential head as this is the only impeller

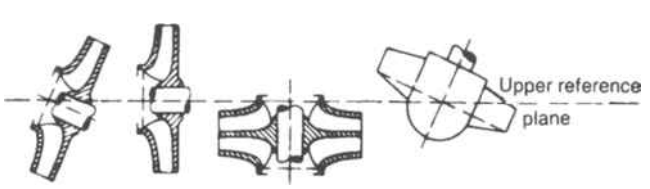

Figure 4.56 NSPH datum for rotodynamic pumps

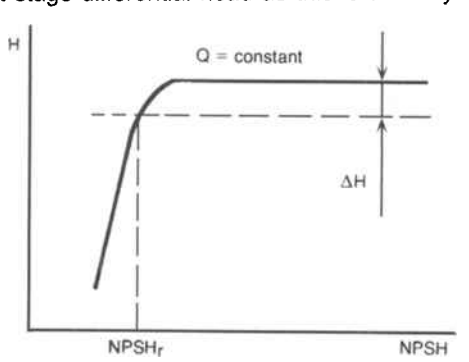

Figure 4.57 Test determination of required NPSH, NPSHr

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which is cavitating. How strictly this requirement needs to be enforced depends upon the size of the pump and how much margin exists between NPSHa and NPSHr.

As mentioned earlier, other NPSHr definitions exist. Cavitation causes damage due to the implosion of the vapour bubbles resulting in a loss of performance and increased power consumption. The concept of damage limitation was studied and a 40000 hour NPSHr criterion has evolved. If effect, the pump manufacturer guarantees that the pump will suffer no loss of performance or require parts replacement due to cavitation damage for 40000 hours of operation. This value of NPSHr is somewhat greater than the 3% criterion and tends to be applied to large pumps and pumps built of costly materials. Obviously, the 40000 hour criterion can not be applied to pumps which will suffer extensive corrosion damage or be eroded by abrasive solids.

Acoustic measuring techniques have advanced during the past few years and the general opinion is that cavitation produces a distinctive noise at about 400 kHz to 500 kHz. This is obviously outside the range of human hearing and detectors are required. Devices are available which include detectors and suitable circuits to initiate alarms or trips. It is possible in the future that a definition will be based on the strength of the cavitation signal.

NPSHr is a function of pump speed. The Affinity Laws can be used for small speed changes. The effect of impeller diameter changes should be referred to the manufacturer.

4.4.3 Permissible suction lift

The permissible suction lift of a pump is controlled by:

- Liquid vapour pressure
- Liquid density
- Pressure on the liquid surface
- Velocity in the pipe
- Friction losses in the pipe
- NPSHr of the pump

Equations 4.38 and 4.39 indicate the mathematical relationship of the variables. P_{min} must never fall below p_v if cavitation is to be avoided. Equation 4.37 can be rewritten in terms of p_v and NPSHr and rearranged to evaluate the limiting value of h_s:

$$h_s = \frac{p_a}{r} - gh_{fs} - \frac{c_s^2}{2} - \frac{p_v}{r} - \text{NPSHr} \qquad \text{Equ 4.40}$$

where p_a and p_v are pressures and NPSHr is in metres.

If the pump is drawing liquid from a vessel which is open to the atmosphere then the maximum suction lift will vary with atmospheric pressure. Atmospheric pressure varies, for example, due to the height above sea level. This is illustrated in Table 4.4. The height of the barometer at sea level is assumed to be 760 mm Hg. Atmospheric pressure also varies according to climatic conditions. At sea level the atmospheric pressure does not normally fall below 720 mm Hg, 960 mbar, 9.8 m H_2O.

Height above sea-level m	Atmospheric pressure Pascal Pa	Millibar mbar	Torr mm Hg	Water column m H_2O
0	$1.013 \cdot 10^5$	1013	760	10.3
1000	$0.899 \cdot 10^5$	899	674	9.2
2000	$0.795 \cdot 10^5$	795	596	8.1
3000	$0.695 \cdot 10^5$	695	521	7.1

Table 4.4 The variations of atmospheric pressure due to height above sea level (760 mm Hg at sea level)

Table 4.4 The variations of atmospheric pressure due to height above sea level (760 mm Hg at sea level)

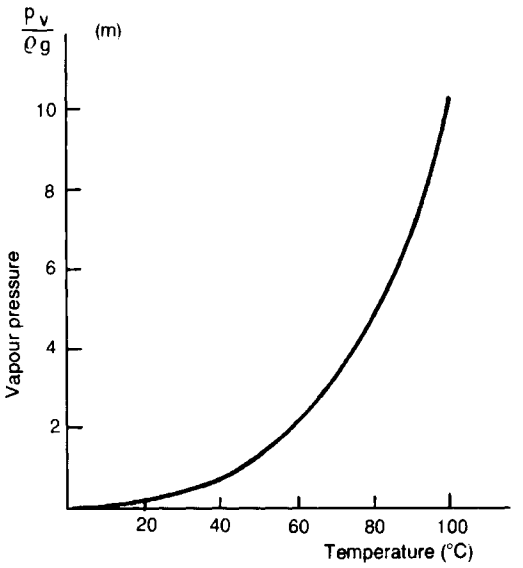

Figure 4.58 Vapour pressure for water

The vapour pressure of a liquid is dependent upon temperature, and Figure 4.58 shows the vapour pressure for water at various temperatures. When pumping warm water or hydrocarbons for example p_v can be large. When pumping boiling liquids the pump must be placed below the liquid free surface so that static head will be greater than the losses in the pipework and the NPSHr. Vapour pressures for a wide selection of liquids appear in Chapter 2, Section 2.6.2.

It should be noted that neither ISO or CEN have approved steam tables for the properties of water. Care must be taken to ensure all interested parties work to the same figures.

The losses in the suction pipe should be kept as low as is practically possible, since they have the effect of reducing the permissible suction lift for cavitation-free operation. The suction line should be of large diameter, short and without unnecessary bends, valves etc. Suction pipe losses and NPSHr vary with flow. It is therefore important to know the minimum and maximum flow at which the pump is to operate and check for adequate NPSHa.

Cavitation and NPSH have so far been explained in the context of rotodynamic pumps. Positive displacement pumps have similar phenomena and criteria although not identical.

Cavitation in positive displacement pumps causes a reduction in flow rather than head or pressure. Any vapour bubbles which form fill part of the displaced volume and reduce the liquid capacity. The bubbles collapse when the pressure increases and flow is reduced by the vapour bubble volume. The imploding bubbles cause material damage as in rotodynamic pumps. Additionally, in pump types such as reciprocating, severe shock waves are produced which can cause fatigue failure in components and bearing failure.

The concept of NPSH becomes NPIP, Net Positive Inlet Pressure, again in two forms, NPIPa and NPIPr. Pressure is used rather than head because all the other pump parameters are expressed as pressure. In positive displacement pump types which are used for process applications handling a wide range of liquids, NPIP is considered to be an equivalent to NPSH; NPIP is quoted in a completely analogous manner. However, some positive displacement pump types are used mostly on one type of liquid and a different approach has become accepted.

Gear pumps and screw pumps are used mostly on lubricating oil. Lube oil has a very low vapour pressure and NPIP testing is difficult due to the high vacuums which must be achieved and attendant air leaks. Also, air and gases evolving from solution tend to mask the true NPIPr value. It has become the practice

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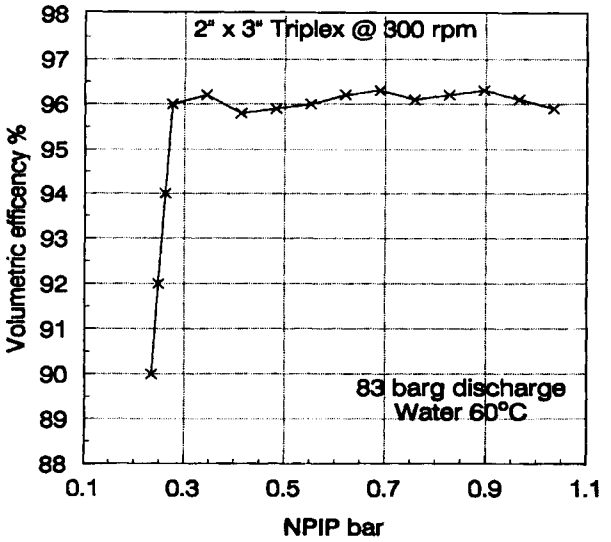

Figure 4.59 Plunger pump NPIP test result

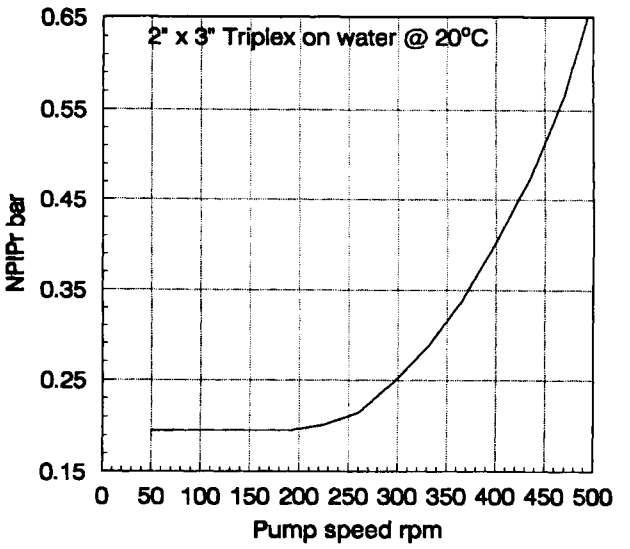

Figure 4.60 Plunger pump NPIP test result

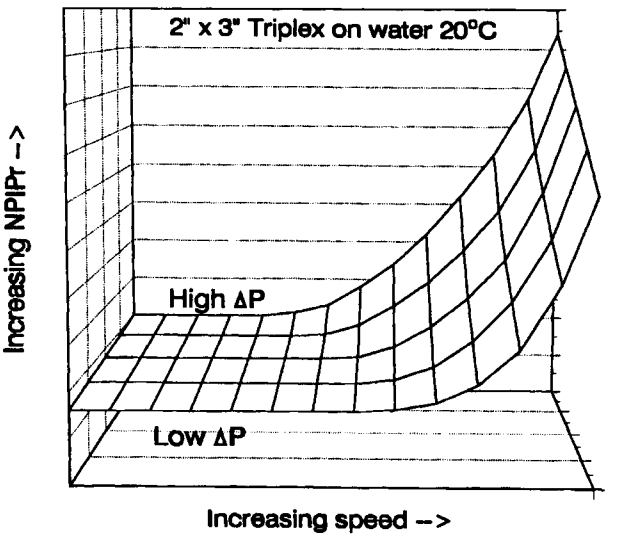

Figure 4.61 NPIPr surface

for these pump manufacturers to specify a minimum suction pressure. Unfortunately some still call this NPIPr and quote values in bara. The picture is further complicated by some positive displacement pump manufacturers who do not fully understand NPSH or NPIP and quote values of NPIPr in bara. It is worthwhile double checking with manufacturers to be sure exactly what figure is being quoted.

Plunger pumps are used extensively for process applications and data is readily available. Pumps are tested at constant discharge pressure while the suction pressure is gradually reduced. When cavitation is initiated, the flow per revolution and the volumetric efficiency are reduced. It is possible that the pump flow may increase slightly due to a very slight increase in speed caused by reduced power consumption. Pumps driven by fixed frequency squirrel-cage motors will run at slightly different speeds depending upon the absorbed power. The flow per rev or volumetric efficiency can be plotted against NPIP, see Figure 4.59.

Reciprocating pumps with plate or plug valves have very distinctive NPIPr characteristics as can be seen from Figure 4.59. Manufacturers of API 674 and 676, reciprocating and rotary positive displacement pumps advise that NPIPr should be specified at 3% flow loss. This is obviously a direct translation of the rotodynamic pump criterion of 3% head loss. 3% flow loss can not be applied universally to reciprocating pumps. Most reciprocating pumps are built with plate valves. The NPIPr characteristic is very clear. NPIPr values should be quoted for 0% flow loss.

Cavitation in reciprocating pumps can produce catastrophic failures due to fatigue. Pumps should not be operated routinely while cavitating. The effects of cavitation on rotary positive displacement pumps should be discussed in detail with the manufacturer, on a case-by-case basis, when deciding on the operating conditions and the purchasing specification.

Reciprocating pump NPIPr characteristics are plotted for constant differential pressure against speed. Some curves may show more than one viscosity or more than one plunger diameter. Increasing viscosity increases NPIPr although the relationship is not easily defined and varies considerably from pump to pump. Increasing viscosity has a more pronounced effect on small pumps than large pumps. Viscosity can also affect the valve dynamics resulting in reduced speed operation. The NPIPr curve does not approach zero at zero speed, see Figure 4.60. A minimum value, depending upon the pump and valve design, is required to avoid cavitation. It may be possible for the NPIPr value to increase at very low speeds, similar to the effect in centrifugal pumps. No test results have been reviewed which show this effect but the possibility should not be ignored if pumps are required to operate below 10% speed.

As explained in Section 4.2.1.5, and shown in Figure 4.15, NPSH can be considered as a 3D surface. The same concept can be applied to reciprocating pumps showing the relationship between speed, differential pressure and NPIPr, see Figure 4.61. The surface shown can be considered as the relationship for a specific viscosity. A higher viscosity would produce another surface, not necessarily of the same shape, above the surface shown. The effect indicated where NPIPr reduces as differential pressure reduces can be completely negated by poor suction valve design.

Rotary positive displacement pumps tend to have a linear NPIPr relationship with speed. The slope of the curve increases as viscosity increases. Usually two or three viscosity curves are shown on one graph to allow interpolation for the required value.

4.4.4 Cavitation effects on pump operation

Cavitation in rotodynamic pumps, if allowed to continue for prolonged periods of time, can cause mechanical damage to the pump. Surface material is removed by the implosion of the vapour bubbles and characteristic damage is visible when inspected. Clearances can be increased causing permanent reduction in performance. Mechanical seals are prone to physical damage. The hydraulic performance is also reduced by the onset of cavitation. The performance is affected in different ways

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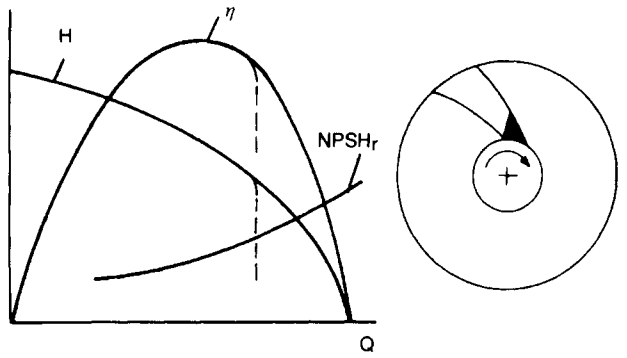

Figure 4.62 Cavitation in a centrifugal pump

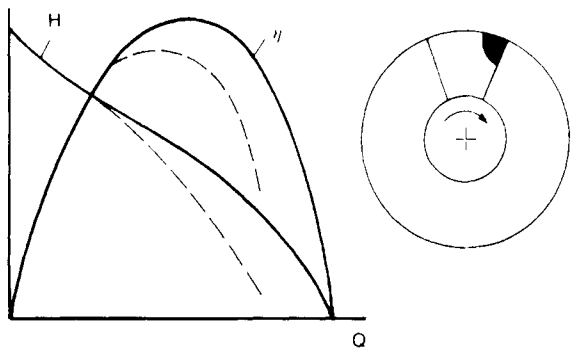

Figure 4.63 Cavitation in an axial flow pump of N_s = 200

depending upon the type of pump. In general, there is an accompanying increase in noise and vibration.

The full lines shown in Figure 4.62 represent the pump's differential head and efficiency for cavitation-free operation. The pump's cavitation sensitivity in the form of NPSHr is also shown. The pump has a low specific speed, N_s = 20. The pump is intended for a system having a certain static suction head. When the flow increases, the NPSHr also increases, while p_{min} decreases. When the positive suction static head is reduced eventually the limits of cavitation-free operation are reached and the pump performance declines rapidly (dashed lines). The rapid decline in performance is associated with the narrow vane passages which are characteristic of pumps having low specific speed. The complete length of the front edges of the vanes are at approximately the same diameter and are subjected to the same velocity. With the onset of cavitation it requires only a slight increase in flow to completely fill the vane passages with vapour. The pump's delivery head can thus be reduced to zero.

The relationship for axial flow pumps is different. Here cavitation occurs on the suction side of the profiles at the tips of the blades where the relative velocity, Δp is greatest. Despite cavitation at the tips of the blades there still remains a large usable amount of blade passage, Figure 4.63. The decline in performance is therefore a more gradual process in the case of axial pumps.

The onset of cavitation in positive displacement pumps causes a reduction in flow rate since part of the displacement becomes filled with vapour. When the vapour bubbles collapse, severe pressure waves can be created which result in fatigue failures. In reciprocating pumps, classic cavitation damage includes:

- Cylinder cracks
- Broken bolts and studs
- Broken valve plates and springs
- High bearing wear

Externally, signs of cavitation include a characteristic "pinking" noise, increased pump vibration and increased pipework vibration due to increased levels of pressure pulsations in both the suction and discharge.

For reciprocating pumps, and some rotary pumps with only a few lobes or teeth, an additional loss, to those already explained, is experienced in the suction pipework. The flow variation caused by the small number of pumping elements requires some additional head to be available to accelerate the flow to keep up with the changing velocity of the pumping elements. This phenomenon has been studied by the Hydraulic Institute in the USA and the German Engineering Federation (VDMA). The extra head required is a function of the:

- Pump type
- Number of pumping elements
- Pump speed
- Velocity in the suction pipe
- Length of the suction pipe
- Nature of the liquid

This extra loss is called Acceleration Head Loss and can be approximated for water, amine, glycol, from the following equation.

$$H_{acc} = 4LUnC \qquad \text{Equ 4.41}$$

where:

H_{acc} = Acceleration Head Loss (m)

L = suction pipe length (m)

U = average velocity in pipe (m/s)

n = pump speed (s^{-1})

C = pump characteristic non-dimensional

The pump characteristic values for single acting reciprocating pumps are:

1 cyl C = 0.4
2 cyl C = 0.20
3 cyl C = 0.07
4 cyl C = 0.11
5 cyl C = 0.04
6 cyl C = 0.06
7 cyl C = 0.03
8 cyl C = 0.045
9 cyl C = 0.025

For other liquids multiply the water value by:

- urea, carbamate, liquids with traces of entrained gases 1.5
- de-aerated water 1.1
- hydrocarbons 0.75
- hot oil 0.6

The approximation given in equation 4.41 must be used with caution. It is, of necessity, an extremely simplistic evaluation of a very complex phenomenon. It was proposed for use on relatively short pipe runs, 15 to 20 diameters. On long runs it tends to be pessimistic, on short runs optimistic. The formula tends to zero as the variables reduce. In practice, Acceleration Head Loss tends to a finite minimum depending upon the pump detail design. If acceleration pressure loss will cause pipe vibration or reduce NPIPa below NPIPr then pulsation dampers may solve the problem. A correctly selected, installed and maintained pulsation damper can attenuate pressure pulsations by 90%.

Positive displacement pumps should always operate with a safety margin between NPIPa and NPIPr. A fixed value of safety margin would be too simplistic and a calculated value is

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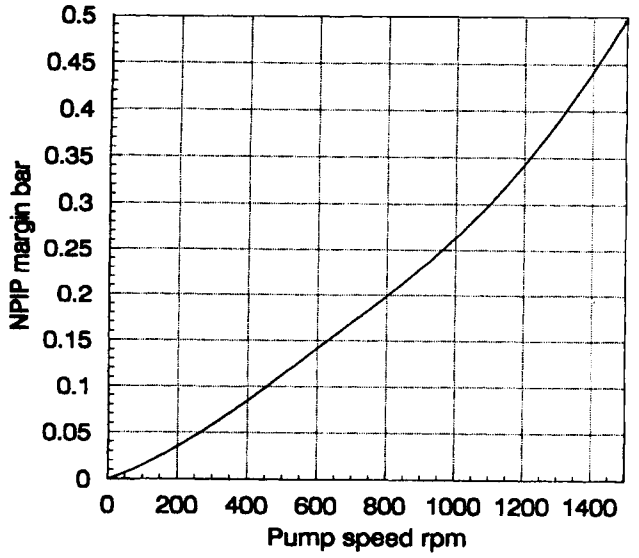

Figure 4.64 Recommended margin of NPIPa over NPIPr for pump speed

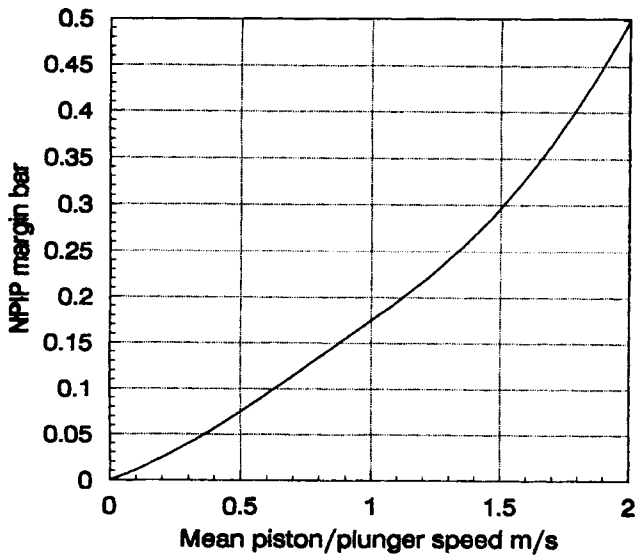

Figure 4.65 Recommended margin of NPIPa over PNPIPr for mean piston/plunger speed

very difficult. Figures 4.64 - 4.66 show guidelines for reciprocating pumps from the British Pump Manufacturers Association.

Values of margins are obtained from both Figures 4.64 and 4.65. A multiplier is obtained from Figure 4.66 depending upon how highly loaded the pump is at the rated conditions and applied to the highest value obtained from Figures 4.64 or 4.65.

4.4.5 Self-priming

A rotodynamic pump operating with air rather than water produces a head rise which is about 1/1000. If the pump is placed above the level of the upper liquid surface and the suction line is filled with air the pump is thus incapable of removing this air by itself when starting. This is referred to as a centrifugal pump being unable to evacuate its own suction line. Before pumping can start it is therefore necessary to fill the pump with liquid, venting and priming. This can be arranged in different ways, see Figure 4.67. Placing the pump below the level of the upper liquid surface, flooded suction, eliminates the problem of evacuation and reduces the risk of cavitation. The NPSH can be achieved for example, by immersing the pump or both the pump and the drive motor, so-called submersible pumps.

Rotodynamic pumps are available in a wide range of designs and configurations. **Not all pumps are self-venting.** Pumps installed as indicated in Figure 4.65 as "dry" may still require venting and priming. Venting and priming after installation or

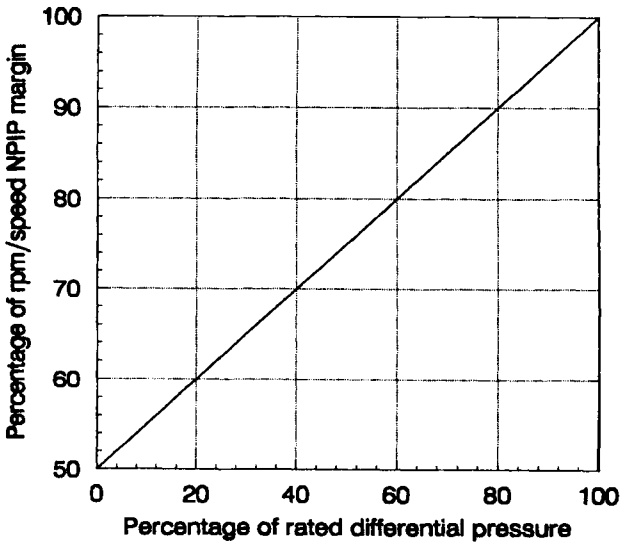

Figure 4.66 Percentage of recommended margin for percentage of rated differential pressure

maintenance, will only be necessary once, unless the liquid level falls below the top of the pump. The manufacturer's operating instructions should be checked to ensure proper pump operation. Submerged pumps are designed to be self-venting and priming.

An alternative method of keeping the pump filled with liquid is shown in Figure 4.68. The non-return (foot) valve prevents liquid draining from the suction line when the pump is stopped. The foot valve is however, always subject to a small amount of leakage and also has the disadvantage that it causes large pressure losses within the suction line when pumping. Provision must be made to fill the suction line initially. The volume of the evacuating tank must be several times greater than the volume of the suction line. It is also necessary to check that cavitation does not occur in the suction line.

The air in the suction line and pump casing can be evacuated with the help of a smaller self-priming pump, Figure 4.69. The priming pump must be connected to the highest point on the pump casing. A wide variety of pump types can be used as evacuating pumps. The most usual are vacuum pumps, side-channel and jet pumps.

All positive displacement pumps are in principle self-priming providing that the inner seals are sufficiently effective and that "dry" running is acceptable during starting. It is therefore advisable to check with the manufacturer with regard to "dry" running in the case of applications having long suction lines. Reciprocating pumps should always be vented and primed prior to starting.

In the case of specially designed self-priming centrifugal pumps the pump is built into a tank containing liquid. The evacuating process is then carried out automatically if the pump is first filled with liquid. The suction line must be dimensioned with regard to both flow losses and evacuation time and must rise continuously towards the pump. Too large a suction line makes evacuation more difficult. A useful rule is to dimension for a suction head loss of 1 to 2 m. If the volume of the suction line above the surface level of the liquid is very large, for example, for suction pipe lengths of several hundred metres, a foot valve must, without exception, be placed below the liquid surface.

4.4.6 Effects of dissolved and entrained gases

The pumped liquid may contain certain quantities of dissolved gas or entrained gas. Lubricating oil, for example, at atmospheric pressure and room temperature can contain up to 10% air by volume, petrol can contain up to 20% air by volume. The ability of a liquid to dissolve gas reduces with reduced pressure and with increased temperature. Gas can therefore evolve in

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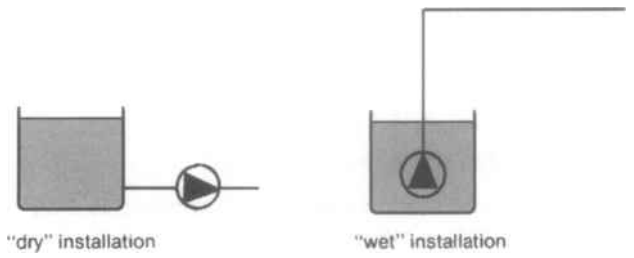

Figure 4.67 Pump installation without suction line evacuation

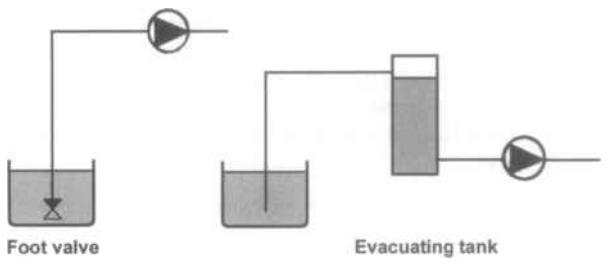

Figure 4.68 Foot valve and evacuating tank

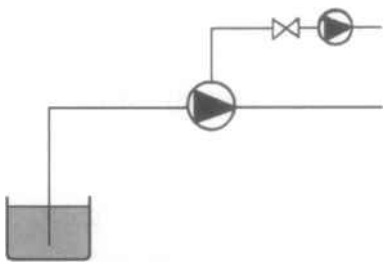

Figure 4.69 Evacuating pump

areas of localised low pressure and re-dissolve when the pressure increases again. This process is similar in some ways to the formation and implosion of vapour bubbles, or cavitation, but is considerably less violent and causes no mechanical damage unless the pressure changes are large, as in positive displacement pumps. When gas separation and cavitation occur simultaneously in rotodynamic pumps the gas has a dampening effect on the implosion process and in this way helps to limit the extent of damage caused by cavitation. Small amounts of air have been introduced into the pump suction on purpose to alleviate certain site problems.

The pumped liquid can also contain entrained gas, usually air. This entrained gas has considerable influence on pump performance. In rotodynamic pumps both the delivery head and the efficiency are noticeably reduced by increased quantities of entrained gas. In positive displacement pumps entrained gas reduces the liquid flow rate and creates pressure pulsations which can cause premature component failure and severe pipe vibration.

As shown in Figure 4.70, the originally stable H-Q curve becomes unstable due to the influence of entrained air. For the system curve shown there are, for a given concentration of entrained air, two alternative intersection points and there is risk of unstable operating conditions.

Certain conditions can cause air to collect in the pump impeller, which can lead to loss of prime. The risk of air collecting is greatest for flows which are less than the design flow. It is normally possible to pump without other difficulties arising, other than reduced efficiency, for air concentrations of up to 2 to 4% by volume measured at the pump intake connection.

Positive displacement pumps usually cope with larger concentrations of air better than centrifugal pumps. For positive displacement pumps operating with a suction lift, the theoretical volume flow is reduced considerably, up to 50%, if the liquid

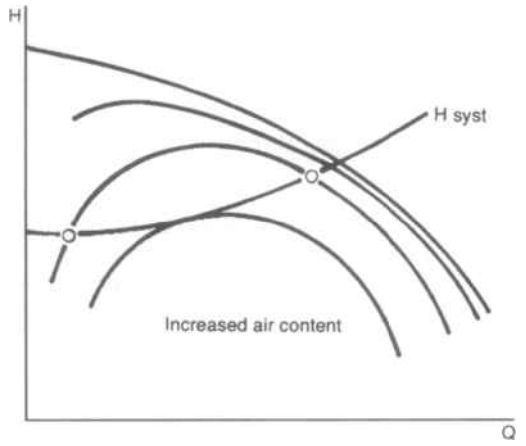

Figure 4.70 The influence of entrained air in water on a centrifugal pump's H-Q curve

contains entrained or dissolved gases. Entrained gas will expand and dissolved gas will evolve and partially fill the displacement volume. Severe pressure pulsations, causing vibration and fatigue damage, may result. Consult pump manufacturers and provide them with full operating conditions, including details of gas contamination, before deciding on a pump type and selection.

Pumps handling cold fresh water from rivers, reservoirs and tanks can suffer damage and short component lives due to entrained and dissolved air. Experience has shown that an allowance must be made for air evolution. The vapour pressure of the water should be increased by 0.24 bar.

Developments in rotodynamic pump technology has resulted in increased gas handling capabilities. Production centrifugal pumps are limited to about 15% by volume before the pump loses prime. Special centrifugal pumps have been able to handle 40% by volume **under controlled test conditions**.

More recent developments in multi-stage axial flow pumps can handle 98% gas volumes for pressures up to 20barg. Special versions of positive displacement pumps have been developed to cope with entrained/dissolved gas continuously. Twin geared-screw pumps can handle 97% gas at pressures up to 40barg. These pumps are also capable of handling small entrained solids, such as sand.

NOTE: It cannot be stressed too often or too strongly that successful pump installations are based on good communications between the user and the pump manufacturer. The pump manufacturer issues guarantees on the basis of operational data provided. If important data is withheld, the guarantee will be invalidated and the manufacturer will not supply replacement parts or service engineers free-of-charge.

The user should not judge which information is important and which is trivial. Forward all information to the pump manufacturer and let the manufacturer decide. This course of action will provide the user with a "water-tight" guarantee. If certain aspects of operation are impossible to quantify, discussions with manufacturers will indicate whether the problem is important or insignificant and if there could be an effect on performance or reliability. It is in the best interests of all concerned to discuss points which need clarification. In general, data sheets for pumps are poor. Full operational conditions are not described in any detail. Filling in an API 610 or API 674 data sheet will not assure a "water-tight" guarantee.

The performance and reliability of many modern pump installations is hampered by poor pipework design. Pumps are not designed to cope with any pipework which the piping designer can fit in. Pumps are designed to operate successfully within an appropriate system. Pump purchasers should ensure the system designer is fully experienced with the types of pumps being

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considered. Pump purchasers do not like buying prototype pumps; but every new pump installation has a prototype piping system. The pump manufacturer will not accept the responsibility for pump failures caused by inherent system defects.

4.4.7 Calculation examples

Example 1

A pump is required to lift water from a river. The minimum barometric pressure is 735 mm Hg and the maximum water temperature is 10 °C. At the rated flow the suction pipe losses, including velocity head and entrance losses, are 1.5 m and the pump is 2 m. What is the maximum suction lift possible?

Water vapour pressure = 0.01227 bara

Water density = 999.8 kg/m^3

Mercury density = 13600 kg/m^3

Atmospheric pressure ≡
735 x 13600 / 999.8 / 1000 = 9.997 m

Water vapour pressure ≡
0.01227 x 100000 / 999.8 / 9.80665 = 0.125 m

Allowance for air evolution ≡
0.24 x 100000 / 999.8 / 9.80665 = 2.448 m

NPSHa = 9.997 - 1.5 - 2 - 0.125 - 2.448 = 3.924 m

Therefore, the maximum suction lift is 3.924 m without any safety margin between NPSHa and NPSHr. If a self-priming pump was used, then the margin for air evolution would not be required and the maximum suction lift would increase to 6.372m.

The density used for water is compressed water data, not saturated liquid data. The water is not at saturated pressure therefore it is compressed.

Example 2

Water at a maximum temperature of 60 °C is to be pumped from an open tank. The minimum barometric pressure is 735 mm Hg and the pump suction will be 3 m below the minimum liquid surface level. If the suction pipe losses, including velocity head and entrance losses, are 4 m at the rated flow what is the NPSHa for the pump?

Water vapour pressure = 0.1992 bara

Water density = 983.2 kg/m^3

Mercury density = 13600 kg/m^3

Atmospheric pressure ≡
735 x 13600 / 983.2 / 1000 = 10.167 m

Water vapour pressure ≡
0.1992 x 100000 / 983.2 / 9.80665 = 2.065 m

NPSHa = 10.167 + 3 - 4 - 2.065 = 7.102 m

Notice that the head equivalent of atmospheric pressure has changed even though the barometric pressure is the same. The water density is reduced because of the increased temperature. Allowance for gas evolution is not necessary because the temperature is high enough to drive off any gas.

Example 3

Water at a temperature of 100 °C is to be pumped from an open tank. The minimum barometric pressure is 735 mm Hg and the suction pipe losses, including velocity head and entrance losses, are 4 m at the rated flow. If the pump has an NPSHr of 5 m at the rated flow, how much static suction head is required to prevent cavitation?

Water vapour pressure = 1.01325 bara

Water density = 958.1 kg/m^3

Mercury density = 13600 kg/m^3

Atmospheric pressure ≡
735 x 13600 / 958.1 / 1000 = 10.433 m

Water vapour pressure ≡
1.01325 x 100000 / 958.1 / 9.80665 = 10.784 m

NPSHa = 10.433 - 4 - 5 - 10.784 = -9.351 m

So the minimum static suction head required is 9.351 m without any safety margin. The head equivalent of atmospheric pressure is increased due to the reduced water density.

Example 4

Water is to be pumped from a closed tank which is pressurised to 4 barg 0.1 barg. The minimum barometric pressure is 735 mm Hg. The maximum water temperature is 150 °C. At rated flow the suction pipe losses are 3 m and the NPSHr of the pump is 3 m, how much positive static head is required to prevent the pump cavitating?

Water vapour pressure = 4.76 bara

Water density = 916.7 kg/m^3

Mercury density = 13600 kg/m^3

Atmospheric pressure ≡
735 x 13600 / 916.7 / 1000 = 10.904 m

Pressure in tank =
(3.99 x 100000 / 916.7 / 9.80665) + 10.904 = 55.287 m

Water vapour pressure ≡
4.76 x 100000 / 916.7 / 9.80665 = 52.949 m

NPSHa = 55.287 - 52.949 - 3 - 3 = -3.662 m

The minimum static head required is 3.662 m without any margin.

Example 5

A three cylinder single acting pump, running at 600 rpm, is to be supplied with water at 60 °C from an open tank. The minimum barometric pressure is 735 mm Hg. The minimum water level above the pump suction is 5 m. The suction line velocity is 0.75 m/s and the line length is 10 m. The suction line losses total 2 m and the pump NPIPr is 0.7 bar. Will the system operate successfully ?

Water vapour pressure = 0.1992 bara

Water density = 983.2 kg/m^3

Mercury density = 13600 kg/m^3

Atmospheric pressure ≡
735 x 13600 x 9.80665 / 1000 / 100000 = 0.980 bara

Static pressure =
5 x 983.2 x 9.80665 / 100000 = 0.482 bar

Friction pressure drop =
2 x 983.2 x 9.80665 / 100000 = 0.193 bar

H_{acc} = 4 x 10 x 0.75 x 10 x 0.07 = 21

H_{acc} ≡ 21 x 983.2 x 9.80665 / 100000 = 2.025 bar

NPIPa = 0.98 + 0.482 - 0.1992 - 0.193 - 2.025 = - 0.955 bar

The system will not work as originally described. The acceleration head loss is too great. A larger suction pipe could be used; this would reduce the friction loss and the acceleration head loss. This would be the best solution for really critical installations. Or a pulsation damper could be fitted; the acceleration head loss could be reduced to 0.2025 bar which would provide an NPIPa/NPIPr margin of 0.867 bar. Figure 4.64 recommends a margin of about 0.15 bar for pumps at 600 rpm. So 90% attenuation of H_{acc} is not necessary for NPIPr purposes. Check the effect of pressure pulsations on pipework vibration before deciding on attenuation required.

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5.1 System curves

The relationship between fixed speed pumps and piping systems is covered by the following sections. Variable speed pumps produce an infinite number of pump characteristics, which can be derived using the affinity laws, and make it possible for a pump to operate virtually anywhere on any system curve.

5.1.1 Pump nominal duty point

When connected in a pipe system a pump will operate at a point of equilibrium between the pump and the pipe system. At the point of equilibrium the energy supplied by the pump is equal to the losses due to resistance in the system. The pump characteristics in this respect are usually represented by the H-Q curve. The equivalent curve for the pipe system is called the system curve and is designated H_{syst}. H_{syst} must include suction **and** discharge losses. Remember both curves indicate differential head.

The flow at which the pump and system curves intersect each other, is the flow which will pass through the pipe system. In order to determine correctly the size and dimensions of the pump and pipe system a knowledge of the characteristics of their respective curves is necessary. This section deals with the characteristics of the pipework system.

Figure 5.1 shows typical curves for a centrifugal pump with a low specific speed, when pumping a water-like liquid. The curves vary in shape when pumping other liquids and when using other types of pumps. Generally, however, the flow is always determined by the balance between the pump and system.

5.1.2 Single pipe system

The system differential head is usually divided into a static component H_{stat} and a loss component h_f.

$$H_{syst} = H_{stat} + h_f \qquad \text{Equ 5.1}$$

The static component, which is generally independent of flow, comprises the difference in static pressures and the level differences between the boundaries of the system. Using the designation as in Figure 5.2:

$$H_{stat} = \frac{p_B - p_A}{r\,g} + h \qquad \text{Equ 5.2}$$

where:

p	=	static pressure (N/m²)
ρ	=	density of the liquid (kg/m³)
g	=	acceleration due to gravity (m/s²)
h	=	difference in elevation (m)

The losses include friction losses in straight pipe and losses in valves and fittings etc.

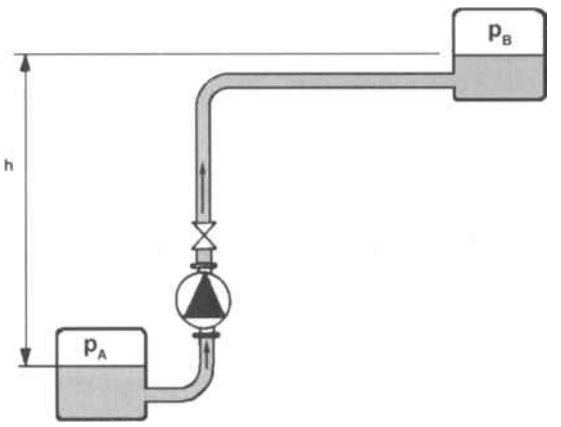

Figure 5.2 Example of single pipe system

$$h_f = h_{(straightpipe)} + h_{(fittingsandvalves)} \qquad \text{Equ 5.3}$$

Using the terms as defined in Chapter 3

$$h_f = \lambda \cdot \frac{l}{d} \cdot \frac{v^2}{2g} + \Sigma\zeta \cdot \frac{v^2}{2g} = \left(\lambda \cdot \frac{l}{d} + \Sigma\zeta\right) \cdot \frac{v^2}{2g}$$

$$= \left(\lambda \cdot \frac{l}{d} + \Sigma\zeta\right) \cdot \frac{Q^2}{\left(\frac{\pi \cdot d^2}{4}\right)^2} \frac{1}{2g} \qquad \text{Equ 5.4}$$

where:

λ	=	loss coefficient for straight pipe
ζ	=	loss coefficient for valves and fittings
Σζ	=	sum of all loss coefficients
l	=	pipe length (m)
d	=	pipe diameter (m)
Q	=	flow (m³/s)
v	=	flow velocity (m/s)

For a given pipework system of l and d, with water-like liquids the loss coefficients λ and ζ are often independent of *Q* for large values of Reynolds Number. The loss of head then becomes approximately:

$$h_f = \text{constant} \cdot Q^2 \qquad \text{Equ 5.5}$$

If the pressures at the system boundaries are such that PA = PB = atmospheric pressure, then H_{stat} equals the elevation difference h, the system differential head then becomes:

$$H_{syst} = H_{stat} + h_f = h + \text{constant} \cdot Q^2 \qquad \text{Equ 5.6}$$

or shown graphically in Figure 5.3.

Note that the system curve represented in Figure 5.3 is based on p_A, p_B and h being independent of the flow Q. It is also as-

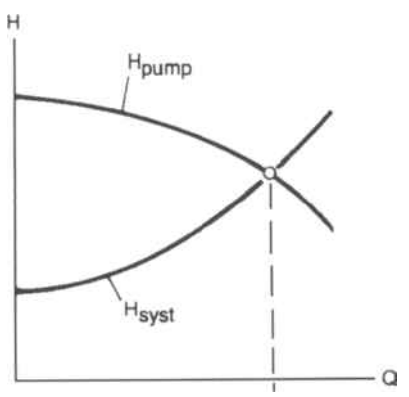

Figure 5.1 Pump nominal duty point

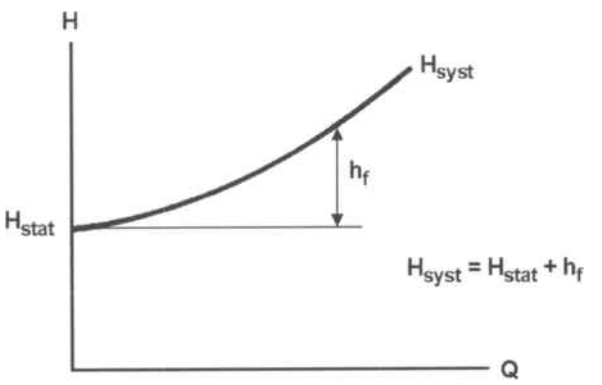

Figure 5.3 System curve

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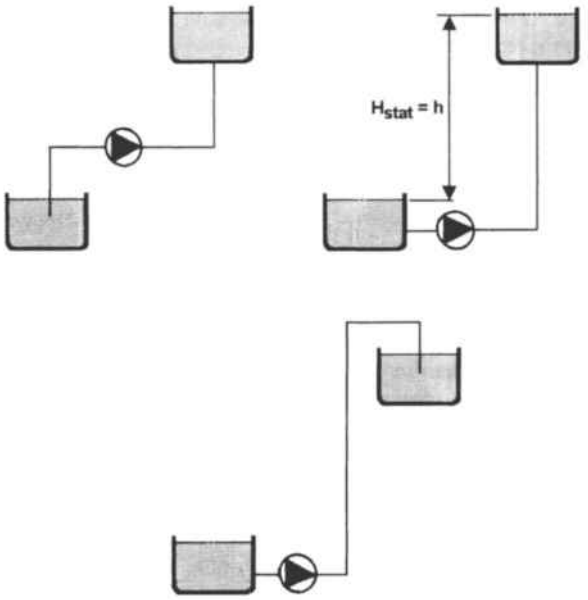

Figure 5.4 Pipework systems with the same static head

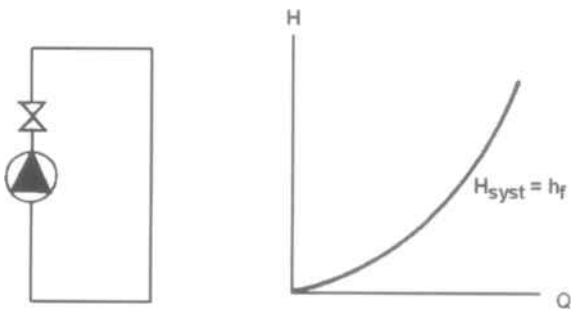

Figure 5.5 Piping system where $H_{stat} = 0$

sumed that λ and ζ are independent of Reynolds Number. These conditions are often, but not always, fulfilled.

Figure 5.4 shows three different pipework systems having the same level differences and therefore the same differential static head. Notice that, from a practical pumping perspective, the systems are different. The first system involves a suction lift whilst the other two have flooded suctions. The physical location of the pump has a significant impact on system design and pump type selection. Non-return valves, essential to prevent reverse flow, are not indicated. In some cases automatic closing isolating valves may be necessary to prevent undesirable forward flow. For certain piping systems, circulating systems for example, $H_{stat} = 0$ and the head of the system consists entirely of pipe flow losses.

For other piping systems with short pipes and considerable pressure or level differences the flow losses are negligible and $H_{syst} = H_{stat}$.

Systems similar to Figure 5.5 are called "pipeline" systems because all the losses are friction losses. Systems similar to Figure 5.6 are called "chemical" systems because it is typical of the arrangements found in chemical and petrochemical installations. When developing the system curve the flow losses are calculated according to Chapter 3.

5.1.3 Variable system curves

Certain conditions can cause the system curve to vary according to the operating situation. Some examples of such situations are shown in Figures 5.7 - 5.11.

When starting up, the discharge pipes are filled with air above the liquid level of the lower tank. The pump must lift the liquid to the highest point in the line. The static delivery head reduces when the line is completely filled with liquid. If the pump's differential head at the zero discharge point, called the "shut-off head", does not exceed H_{stat} at the start then pumping cannot begin.

For installations similar to Figure 5.8 the static head varies as the level difference between the liquid surfaces in the vessel changes. Similar system curve changes occur in enclosed vessels, in which the pressure is dependent upon the operational situation.

The system curve's loss component can also vary according to the operating situation. The most usual case being when rust or other deposits begin to build up in a pipeline after a period of operation. This causes an increase of flow resistance in the pipe-

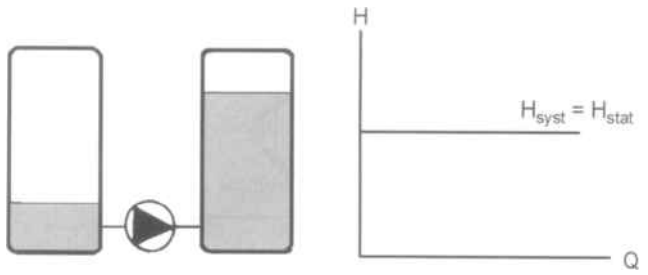

Figure 5.6 Piping system where $h_f = 0$

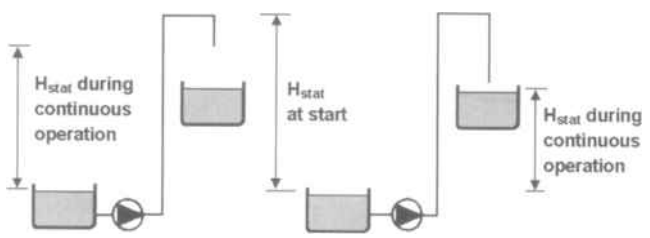

Figure 5.7 Static head at start and during continuous operation

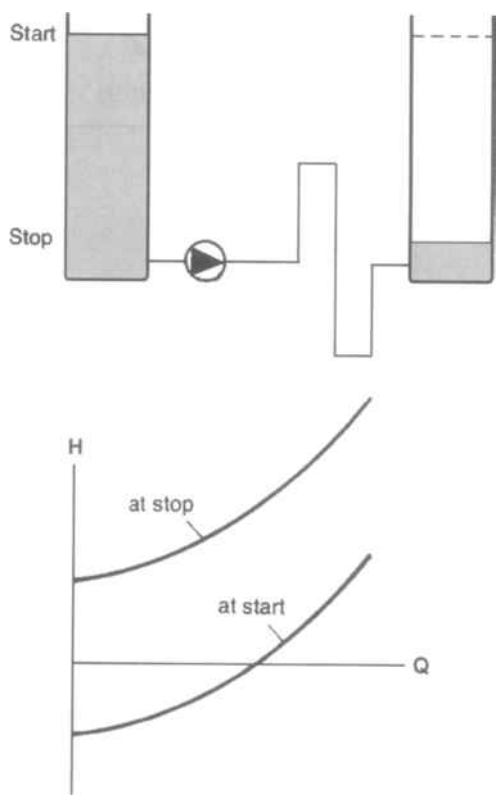

Figure 5.8 System curve with varying static head

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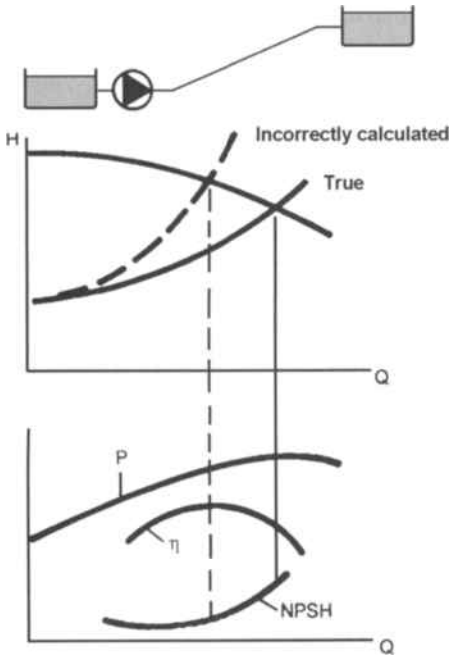

Figure 5.9 Consequences of incorrectly calculated pipe losses

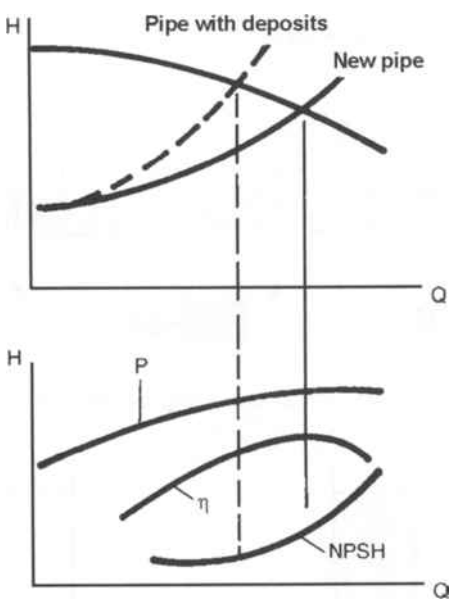

Figure 5.10 The effect of deposits on pipes

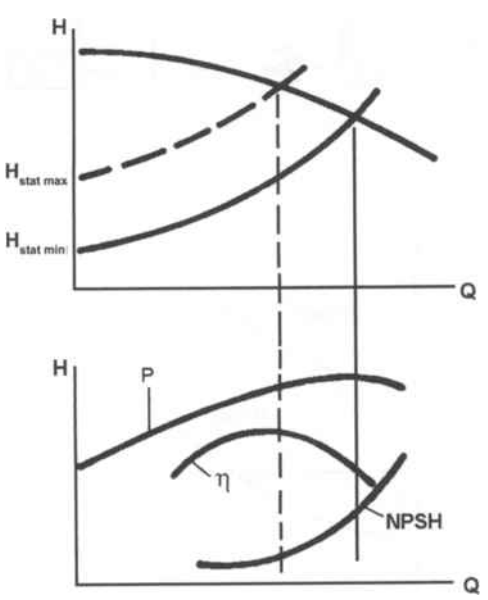

Figure 5.11 The effect of varying static head

line which results in a steeper system curve. Other examples of varying loss curves are:

- Temperature variations
- Viscosity
- Reynolds Number
- Concentration variations in the case of liquid-solid mixtures
- Dosage variations etc.

An intentional change to the system curve is brought about by throttle-regulation, and discussed in Chapter 6.

The pump H-Q curve is also affected by viscosity and wear for example, see Chapter 1.

The relevance of the system curve to operating situations can be illustrated by some examples:

The pump installation shown in Figure 5.8 has been designed to be "on the safe side", the pipe losses having been estimated too high. The actual system curve results in a larger flow than required, which means greater pump power requirement and reduced efficiency, see Figure 5.9. This situation gives rise to the risk of pump overload and cavitation. The desired flow can be maintained by introducing a flow resistance into the piping system.

Figure 5.10 illustrates the reduction of pump flow caused by the build up of deposits (scale, rust, etc.) in the pipeline. Since the size of pump has been chosen in order to obtain optimum efficiency for operation with new pipes, the subsequent flow reduction results in reduced pump efficiency.

The pump in Figure 5.11 has been chosen to suit a particular operational situation, which is equivalent to $H_{stat,\ max}$ i.e. when the liquid levels in the lower and upper vessels are lowest and highest respectively.

In the alternative case $H_{stat,\ min}$ a larger flow is obtained. When plotting pump and system curves, it is not sufficient to plot only the differential head. The NPSHa and the NPSHr should be plotted for the various flows considered. If allowances are added to the system losses, for eventual fouling for example, the operating conditions for the new pipe must be checked if balancing valves are not fitted.

If the plotted characteristics indicate that there is no pump available to suit the system NPSHa, the "system" must be modified. In this context, the system includes some operating conditions. In order to provide more NPSHa the suction pressure must be increased or the difference between the suction pressure and the vapour pressure must be increased. The following courses of action should be considered:

- Re-route the suction pipe to make it shorter
- Move the pump closer to the suction source and reduce the length of the suction pipe
- Lift the suction source up to increase H_{stat} on the suction side
- Increase the suction pipe diameter
- Reduce the liquid temperature to reduce the vapour pressure

5.1.4 Branched pipe systems

The first example of a branched pipe system is a circulation system, see Figure 5.12. In such a case $H_{stat} = 0$. The flow Q_p, which passes through the pump, is divided at the branch point. Continuity dictates that

$$Q_p = Q_A + Q_B \qquad \text{Equ 5.7}$$

Both branches A and B have their own system curves, the sum of which determines the resultant system curve. The intersec-

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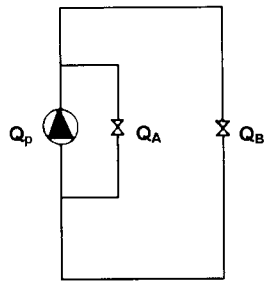


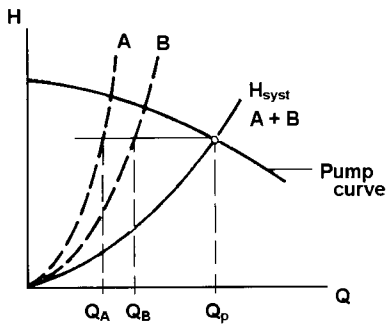
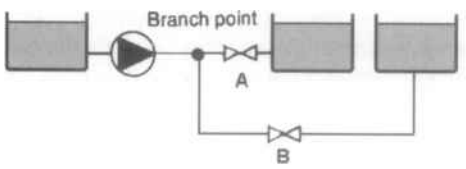


Figure 5.12 Branched circulation system



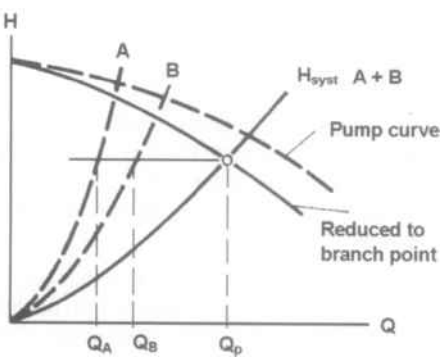


Figure 5.13 Branched pipe system $H_{stat} = 0$

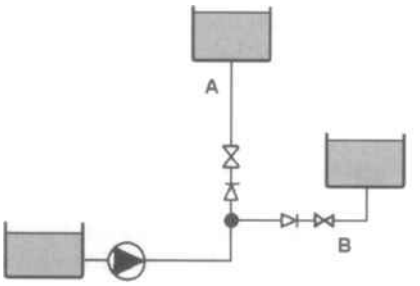
tion point of the resultant system curve and the pump curve determines the pump's operating point.

Figure 5.12 also shows how large a portion of the pump flow flows through each respective branch. The division of flow depends upon the magnitude of the losses in each respective branch. In this example it is assumed that the branch point is near to the pump i.e. that flow losses between the pump and the branch point can be neglected.

The next example, Figure 5.13, also takes into consideration the losses in the main pipeline up to the branch point. This is done by reducing the pump curve by the magnitude of the losses up to the branch point. The "reduced" curve is then matched to the remaining system curve as before.

In the third example as shown in Figure 5.14, the pump curve is first reduced to the branch point. Thereafter the system curves A + B are added to the resultant system curve H_{syst} A + B. The intersection point between the reduced pump curve and the resultant system curve determines the pump's operational point.

In the fourth example as shown in Figure 5.15, there is a level difference, namely a supply head between the supply vessel on the suction side and the branch point. The system curve for the



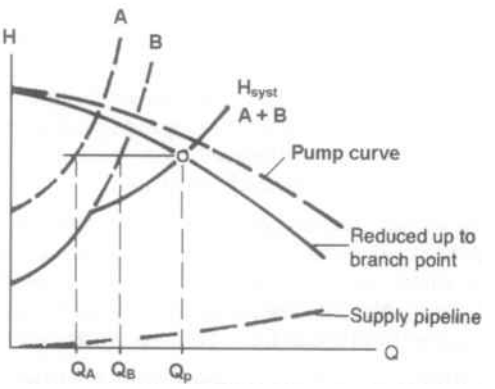
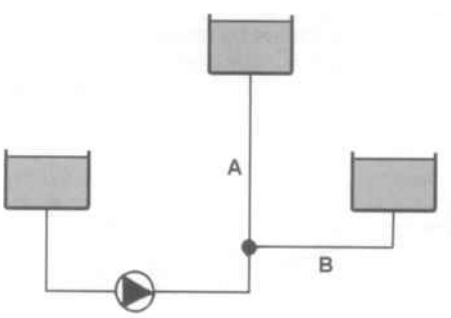


Figure 5.14 Branched pipe system with static differential head

supply pipeline exhibits therefore, a static head, $H_{stat} < 0$. The method is the same as in the previous examples. First reduce the pump curve with the supply line curve up to the branch point. Then determine the system curves for the branches A + B from the branch point. The resultant branch curve is then matched to the pump's reduced curve.

In principle, this method can be applied to calculate any pipework system, however complicated and containing any number of branch points. The first step is to determine the resultant sys-



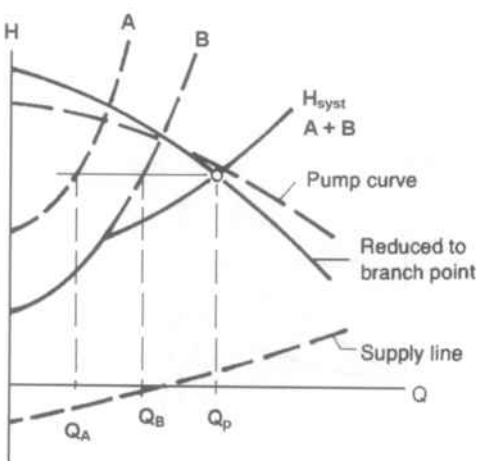


Figure 5.15 Branched pipe system with static supply line

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tem curve for the branch furthest from the pump in relation to the furthest branch point. The next step is to deal with the next branch point and so on until the branch point nearest to the pump is reached.

Remember the system curve is a combination of suction and discharge losses. The H for the vertical axis in Figures 5.1, 5.3, 5.5, 5.6, 5.8 and 5.9 to 5.15 is really ΔH, the losses from the beginning of the suction pipe to the end of the discharge pipe. Rotodynamic pump curves indicate ΔH across the suction and discharge connections of the pumps.

A common problem is the sizing of a piping network with many branches and many tapping points in such a way as to guarantee a predetermined flow and pressure at each tapping point. Sizing is carried out so that the pipeline requiring the most pressure, usually the pipeline to the furthest tapping point, is dimensioned first. The branch lines are then calculated on the basis of the excess pressure available at the branch points. The excess pressure at the branch points can be such that it is necessary to equip a branch line with extra throttling to prevent excessive flow and unbalancing the system.

Throttle valves can be fitted to balance a complex system. As was shown in Figure 3.25, some applications have well defined velocity restrictions depending upon the quality of the pipework construction. Also, throttle valves can be used to correct imbalance caused by irregular fouling of different parts of the system.

5.1.5 Viscous and non-Newtonian liquids

In the earlier figures, the pump and system curves have been representative of the relationships for centrifugal pumps and water-like liquids. For viscous and non-Newtonian liquids the curves have different shapes.

For viscous liquids the laminar flow regime is maintained at higher flow velocities. The system curve thus becomes modified. The centrifugal pump H-Q curve falls off more quickly with increased flow for higher viscosities.

High viscosity liquids are normally pumped by positive displacement pumps. Rotodynamic pumps can handle higher viscosity liquids by derating the pump, usually reducing the speed to reduce the torque required. A common bench-mark used to separate rotodynamic pumps, low viscosity, from positive displacement pumps, high viscosity, is 300 cSt. This is not an absolute number; it depends upon the size of the pump and can be exceeded when circumstances necessitate using a rotodynamic pump. The high viscosity tends to reduce the internal leakage, slip, of rotary positive displacement pumps making them more efficient. High viscosity causes higher pipe losses. The system curve becomes completely linear if laminar flow is maintained at the duty conditions. The increased flow and increased pressure means greater power requirements. See Figures 5.16 and 5.17.

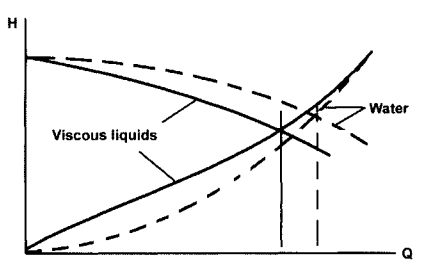

Figure 5.16 Rotodynamic pump and system curves for more viscous liquids compared to water

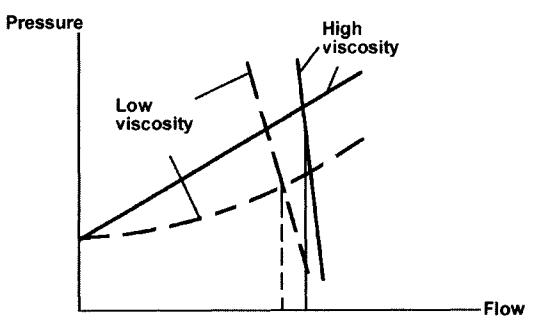

Figure 5.17 Rotary positive displacement pump and system curves for different viscosities

The pumping of chemical paper pulp is an example of system curves for non-Newtonian liquids, see Figure 5.18. The liquid characteristics of pulp suspensions is reflected in the shape of the system curve. The centrifugal H-Q curve also changes in relation to that for water.

Since the pump's operating point is always determined by the equilibrium between the pump and system, the shape of these curves is significant from the point of view of pump installation performance. This applies also to the sensitivity to disturbances and part-load characteristics.

5.2 Pressure drop across valves

5.2.1 General

For pump installations which use valves as a means of controlling the flow, the pressure drop across the valves may constitute a considerable proportion of the piping system head losses. The system curve which is the basis for the determination of the required differential head for the pump, cannot be defined with sufficient accuracy if the pressure drop across the valves, particularly control valves, is not considered. The following considerations apply to the choice of valves and the pressure drop across the valves for the purpose of determining the required pump performance. A more detailed treatment of flow control is to be found in Chapter 6, Flow regulation.

5.2.2 Isolating valves

Isolating valves are those valves which are intended to be either fully open or fully closed. Isolating or stop valves are installed in an installation in order to make components accessible for service; to direct flow in another direction and for other similar functions.

The key requirements of an isolating valve are low pressure drop in the open position and good leak-free sealing in the shut position. Information regarding the calculation of pressure drop in the fully open position is given in Chapter 3. The most reliable method, however, is to obtain the value of the loss coefficient λ or flow coefficient K_v from the valve manufacturer.

5.2.3 Non-return valves

Non-return or check valves are installed in a pipe system in order to prevent the reverse flow of liquids. There are many different types of non-return valves available, and it is normal to select the one which gives least pressure drop. In some cases it is necessary to have exceptionally fast-closing check valves. These are biased, by means of spring loading for example, and give rise to large pressure drops. These valves can also give rise to tremendous pressure surges due to the water hammer effects they create. Very slow closing valves are available specially designed to eliminate water hammer. Since the effective flow area varies with the flow through the valves, the loss coefficient ζ also varies with flow in non-return valves. Pressure drop calculations are best made by using data supplied by the valve manufacturer.

5.2.4 Control valves

The purpose of control valves, in contrast to the other valve groups, is to create head losses and thereby regulate the flow in the pipework system. The choice of valve is therefore made on the basis of entirely different criteria than for isolating or non-return valves.

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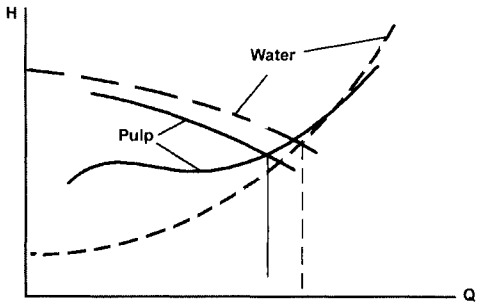

Figure 5.18 Rotodynamic pump and system curves for water and paper pulp

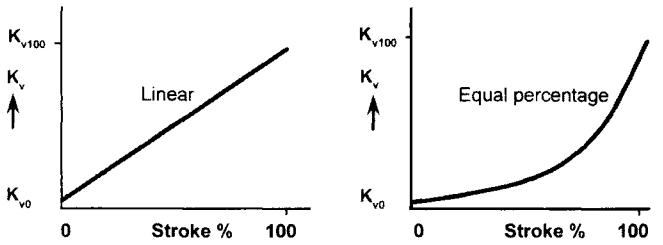

Figure 5.19 Linear and logarithmic control valve characteristics

The principal requirements of a control valve are based upon its regulating characteristics. These characteristics are also dependent upon the interaction between the pump, control valves, pipe system and other control equipment. The valve can only regulate flow by increasing the liquid velocity locally to create a high pressure drop. The capability of the valve to regulate ceases when the pressure drop across the valve approaches zero. If it is required to sustain regulation capability even at Q_{max}, maximum flow in the proposed process, the valve must have an available pressure drop at that flow which is somewhat greater than that for the fully open valve position. Control valves are sized so that they are never fully open or closed; a 10% margin of travel at each end of the stroke is a minimum safety margin.

The design of other control equipment is simplified if the flow through the pipe system has fairly linear characteristics in relation to the position of the valve. This requirement also makes a large pressure drop across the valve desirable at Q_{max}.

The flow through a valve is calculated using the formula:

$$Q = K_v \sqrt{\frac{\Delta p}{\rho}} \qquad \text{Equ 5.8}$$

where:

Q = flow (m^3/s)

K_v = valve flow coefficient (m^2)

Δp = pressure drop across the valve (Pa)

ρ = density of the liquid (kg/m^3)

A measure of the valve capacity and size is its flow coefficient A_{v100} at the fully open position, 100% stroke. To make a rapid assessment of a suitable valve size the following rule of thumb is used:

$$K_{v100} \geq \frac{Q_{max}}{\sqrt{g\,(0.1 \cdot H_{stat} + 0.3 \cdot h_{fpipe})}} \qquad \text{Equ 5.9}$$

K_{v100} = flow coefficient at fully open position 100% stroke (m^2)

Q_{max} = maximum flow with regulation requirement (m^3/s)

g = acceleration due to gravity (m/s^2)

H_{stat} = system static head (m)

h_{fpipe} = total head loss for pipe system at Q_{max} excluding valve losses (m)

If the two sides of the equation are equal then this indicates that the valve at full stroke would just precisely allow the flow Q_{max} to pass through and constitutes a theoretical lower limit of valve capacity.

The given pressure drop at Q_{max} is a compromise between, achieving good regulating characteristics for the system without using too large a valve on the one hand, and not necessitating too high an additional pump differential head with associated pump and energy costs on the other. Control valves are not sized by the pipe size. A control valve will frequently be two sizes smaller than the pipe.

The next step is to select the valve characteristic. The two most common are linear and logarithmic, equal percentage. In principle a valve can be given any characteristic by specially shaping the internal components, the valve trim i.e. plug etc. The valve is described as having a linear characteristic if its flow coefficient increases in direct proportion to the stroke. If K_v increases logarithmically in relation to the stroke then the valve is referred to as having an equal percentage characteristic.

Between these two types are valves with quadratic, V-port characteristics. Many valves display characteristics which differ to a greater or lesser extent from these mathematically defined forms.

The following rule of thumb is used as a guide when selecting a suitable valve characteristic:

If the relationship between the valve pressure drop at Q_{min}, minimum flow which is to be regulated, and at Q_{max} is less than 3, try a valve having a linear characteristic. If $\Delta pQ_{min}/\Delta pQ_{max} > 3$, choose a logarithmic characteristic.

With a lower limit of K_{v100} and with a preferred valve characteristic, a preliminary choice of control valve can be made. Alternatives to A_v are used in practice:

$$K_v = \frac{Q}{\sqrt{Dp \cdot \frac{rH_2O}{r}}} = \left[\frac{m^3/h}{bar^{1/2}}\right] = \frac{K_v \cdot 10^6}{28} \qquad \text{Equ 5.10}$$

$$C_v = \frac{Q}{\sqrt{Dp \cdot \frac{rH_2O}{r}}} = \left[\frac{US\ gpm}{psi^{1/2}}\right] = \frac{K_v \cdot 10^6}{24} \qquad \text{Equ 5.11}$$

In catalogues, the specified value for K_v and C_v are derived by testing with water at 5 to 30°C, which must be considered when selecting a valve for high viscosity or non-Newtonian liquids.

Having made a preliminary selection of pump and control valve and after calculating the system curve excluding the valve, it is possible to check the pump installation regulating characteristics.

5.2.4.1 Calculation example

For a particular pump the system curve, excluding the valve, is determined, using Figure 5.20 upper illustration. The flow is to be valve regulated within the range 0.03 to $0.1 m^3/s$. Select the valve and pump size and check the regulating characteristics of the pump installation.

At maximum flow with regulating requirements:

$Q_{max} = 0.1\ m^3/s$, $H_{stat} = 18$ m, $h_{fpipe} = 20$ m

According to the rule of thumb, Equ 5.9, at Q_{max}

$h_v = (0.1 \cdot 18) + (0.3 \cdot 20) = 7.8$ m

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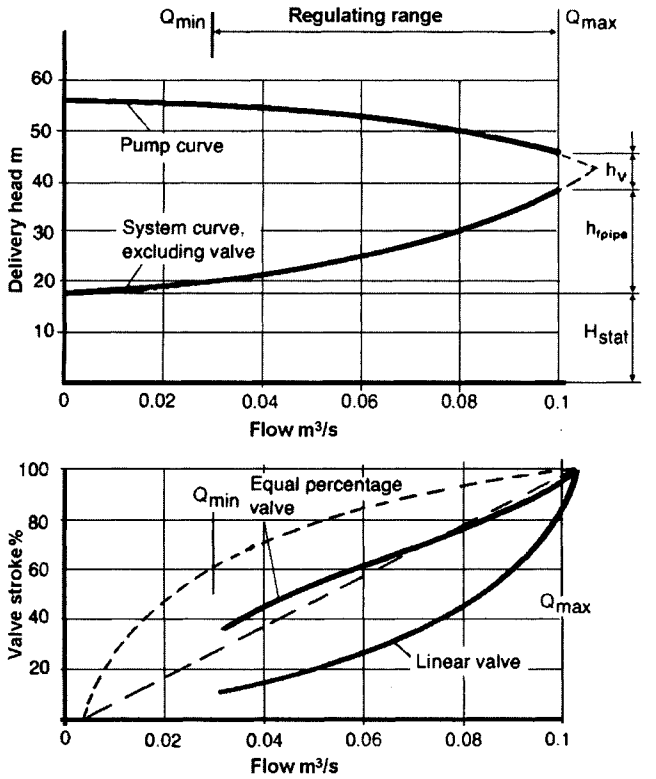

Figure 5.20 Evaluating the regulating characteristics of the pump installation

Select a pump whose H-Q curve passes through the point, H = 18 + 20 + 7.8 = 45.8 m at Q = 0.1 m³/s.

According to the rule of thumb, Equ 5.9,

$$K_{v100} \geq \frac{0.1}{\sqrt{9.81 \cdot 7.8}} = 0.0114$$

From the pump and system curves the largest and smallest pressure drop across the valves within the regulating range can be read. The relationship between these is 35/7.8 = 4.5 which is > 3 and indicates a logarithmic valve characteristic. Choose a regulating valve with K_{v100} = 0.013 having a logarithmic characteristic. Calculate the valve setting, percentage stroke, at various flows by using the formula:

$$Q = K_v \cdot \sqrt{g \cdot h_v}$$

$$K_v = K_{v0} \cdot e^{\frac{s}{100} \cdot \ln\left(\frac{K_{v100}}{K_{v0}}\right)} \qquad \text{Equ 5.12}$$

where:

s = actual setting (stroke) (%)

K_{v0} = flow coefficient at s = 0 (m²)

The calculated result is shown by the lower illustration in Figure 5.20.

In the example K_{v0}= 0.04 · K_{v100}, i.e. the valve regulating range is 1:25. For comparison the pump installation regulating characteristics have also been calculated for a valve having linear characteristics. For the linear valve:

$$K_v = K_{v0} + (K_{v100} - K_{v0}) \cdot \frac{s}{100} \qquad \text{Equ 5.13}$$

The largest flow which can pass through the pump installation is 0.102 m³/s and corresponds to the valve fully open condition. Within the regulating range the valve stroke varies between 35 and 96% of full stroke for the logarithmic valve. Corresponding figures for the linear valve are 87 and 9%. The logarithmic valve gives the installation an approximately linear characteristic.

The chain-dotted curves in the lower illustration in Figure 5.20 show the installation's regulating characteristics at constant pressure drop across the valve, i.e. without considering the effects of the pump and the shape of the system curve.

Note that the pump data selected in the example neglects tolerances of the installation components, which is not to be recommended in practice.

5.2.4.2 Summary

- The pressure drop across control valves can be considerable and cannot be neglected when determining pump operating characteristics.
- The regulating characteristics of the installation determines the pressure drop required across a control valve.
- The smaller the valve pressure drop in relation to the pipe system losses, the more distorted the valve characteristic.
- Valve pressure drop must be "paid for" with increased pump power and can generate considerable energy costs.
- Rules of thumb can never be applied universally.
- All components in the pump installation are subject to certain hydraulic tolerances. Deviations from the nominal performance for pumps, control valves and other fittings, together with inaccuracies when calculating pipe losses etc. are unavoidable and must be given careful consideration. In valve regulated systems the control valve is the component which is used to compensate for the component tolerances.

5.3 Multiple pump systems

5.3.1 Series pump operation

Series pump operation is used when high differential heads are required. Series operation is especially applicable for high differential heads in combination with comparatively small flows. By dividing up the differential head between more than one pump, a higher specific speed, N_s, is maintained for each individual pump, thereby increasing efficiency.

Multi-stage pumps are a compact, integral form of series connection. Systems incorporating a number of pumps, where the pumps are situated in close proximity to one another are simply dealt with by calculating a common performance curve for the complete "pump package". The common pump curve is then compared with the system curve.

For series operation the resultant H-Q curve is obtained by adding the actual differential heads for each value of flow, Figure 5.22.

The resultant efficiency at various flows is calculated from the relationship:

$$h = \frac{r \cdot g \cdot Q\,(H_1 + H_2)}{P_1 + P_2} \qquad \text{Equ 5.14}$$

When two similar pumps are connected in series the same efficiency curve applies for the "package" as for each separate pump unless the liquid is very compressible. For very compressible liquids, liquified gases are the worst, the flow throw

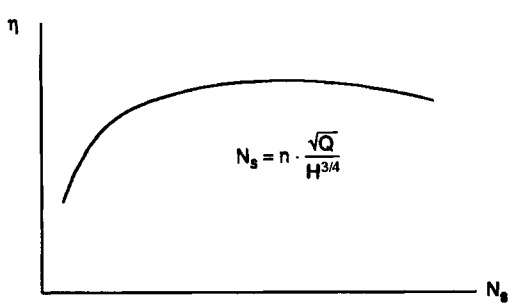

Figure 5.21 Maximum efficiency for centrifugal pumps at various specific speeds

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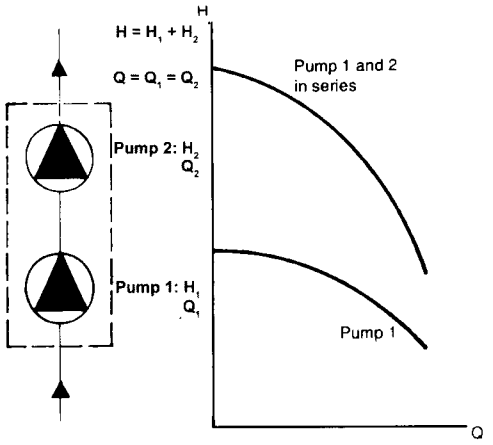

Figure 5.22 H-Q curve for series pump operation

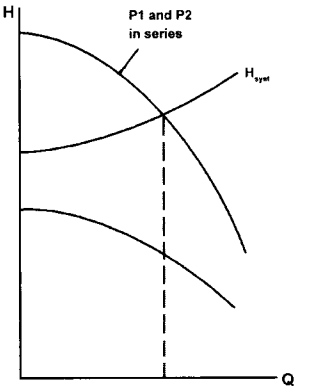

Figure 5.23 Pump nominal duty point for series operation

the second pump may be considerably smaller than the first if the pump differential head is high.

The pumps P_1 and P_2 shown in Figure 5.23 cannot operate individually on the system because one pump does not develop sufficient differential to overcome H_{stat}. Depending upon the pump type and size, precautions may be needed to prevent one pump running without the other. The second pump may cavitate if the first pump is not running and both pumps may overheat. Instrumentation and control logic should be fitted to protect the pumps. It is possible to pump through a rotodynamic pump. The head loss through a pump is proportional to the specific speed. As the liquid flow through a pump increases it will begin to rotate in the normal direction. The liquid and impeller angles will match and the head loss will not rise as steeply. The pump should not overspeed.

For most liquids, the same flow passes through both pumps. The inlet pressure for P_2 is the pump outlet pressure of P_1. The pump pressure rating and shaft seals can therefore be affected. The axial thrusts, which are dependent on the differences between the internal and atmospheric pressure, will also be different for the two pumps. For low differential pressures, identical pumps may be used. As the differential pressure increases, the second pump may require significant modifications to accommodate the increased inlet pressure.

One very common occurrence of series operation is used to overcome the problem of low NPSHa discussed in Section 5.1.3. Some rotodynamic pumps are required to produce very large differential heads. Even with multi-stage pumps, 12 stages are common, the ΔH at normal motor speeds is not sufficient. The pumps are driven at higher speeds by step-up gearboxes or by steam/gas turbines. The NPSHr of the pump increases.

To overcome the problem, and to avoid very costly solutions such as raising the suction source, a low power booster pump is operated in series with the main pump. The function of the booster is to supply NPSHa only. The differential head is therefore very small compared to the main pump and the power absorbed is low; efficiency is not really a consideration.

The booster pump would generally be a single stage pump, perhaps double entry, running slowly to match the NPSHa. If the main pump is geared up from an electric motor, the booster pump may be geared down from the same motor. If the main pump is turbine driven, the booster will probably be motor driven. In this type of series pump installation the pumps will be completely different, reflecting their respective functions.

Booster pumps are also used on positive displacement pumps. Depending upon the nature of the liquid the booster pump may be another positive displacement pump or a rotodynamic pump. High pressure positive displacement pumps working

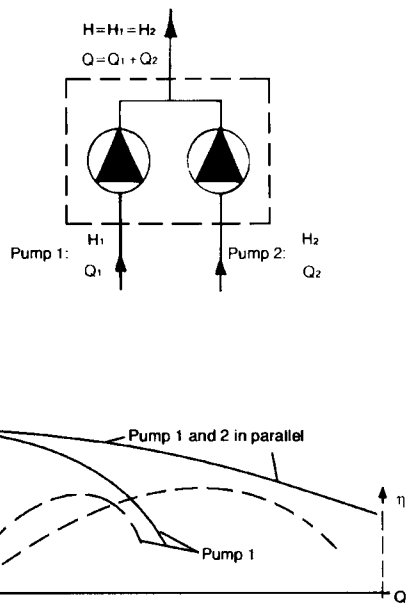

Figure 5.24 Parallel pump operation

with water-like liquids may use a centrifugal pump booster to provide adequate NPIPa.

Pumps handling more viscous liquids would use a low pressure positive displacement pump. When two positive displacement pumps are used, the booster should be oversized slightly to allow for wear. A back-pressure regulator and a relief valve should be fitted to the inter-connecting pipework to prevent excessive pressures. It is also a good idea to fit a pulsation damper, with extra gas volume, to reduce any pulsation levels and function as an accumulator.

5.3.2 Parallel pump operation

For parallel pump operation, the resultant H-Q curve is obtained by adding the actual flows for each head value, Figure 5.24.

Parallel pump operation is used to achieve large flows and deal with greatly varying flows by operating a varying number of pumps. The effect of parallel operation is dependent largely upon the shape of the system curve.

For steep, friction dominated system curves only a small increase in flow is achieved by parallel operation. A much better effect is obtained if the system curve is dominated by static head.

The pump's specification, Q and H, is based upon the pump's nominal duty point, B in Figure 5.25. If only one pump operates, the duty point will then be at point A. This can cause conditions where there is risk of cavitation, because of the lower NPSHa value at point A than at B, and risk overloading of the motor.

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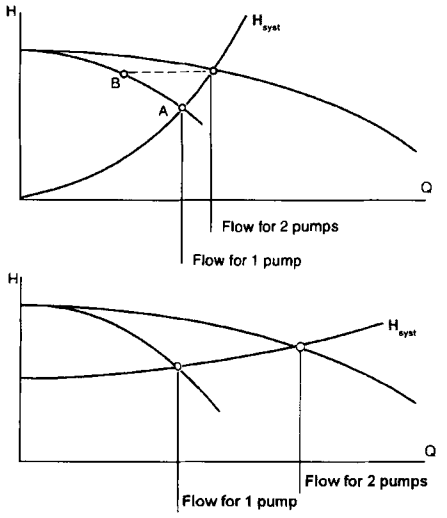


Figure 5.25 Parallel operation of 2 similar pumps with different system curves

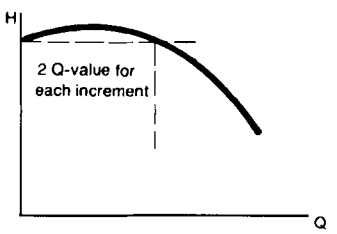


Figure 5.26 Unstable pump curve

The resultant efficiency curve for the "package" is calculated from the relationship:

$$\eta = \frac{r \cdot g \cdot H\,(Q_1 + Q_2)}{P_1 + P_2} \qquad \text{Equ 5.15}$$

For two similar pumps the efficiency values are the same for each separate pump except that the Q values are displaced by a factor of two, Figure 5.25.

Unstable pump and shallow pump curves should be avoided when pumping in parallel. In any case the intended operational range should not fall within the unstable region without taking special precautions. Special minimum flow valves or orifices can be fitted to prevent the pump reaching the peak of the curve. See Figure 5.26.

Typical problems which can occur include irregular flow due to varying operational points and the inability of a pump to open it's non-return valve when starting. Non-return valves are fitted to prevent reverse flow through stationary or standby pumps. Characteristic curves for positive displacement pumps are generally so steep that parallel operation causes no problems. Different types of pump can be operated in parallel to achieve the best operating efficiency at all flows.

5.3.3 Pressure boosting

Pressure boosting pumps are primarily employed in long pipelines and large circulation systems. A common problem is that of pressure in a distribution network being insufficient for certain higher locations, for example, the water supply to high-rise apartment blocks; or when an additional tapping point is connected to an existing network.

Pressure boosting is a form of series connection where the pumps are located at considerable distances from each other. The advantages of pressure boosting are:

- A better distribution of the specified pressure in the pump installation is achieved
- Pump and energy costs can be reduced, (applies to branched pipework systems), see Figures 5.27 and 5.28

In Figures 5.27 and 5.28, higher pressure rating may be required for the upstream section of the pipeline and pump for Alternative I, or for the second pump in the event that the pressure boosting for Alternative I is also divided into two pumps.

Branched pipework systems are usually sized by calculating for the pressure requirements of the furthest tapping point. The required pump specification is therefore based according to this pressure requirement and the total flow consumption. The application of pressure boosting principles can provide an alternative, as illustrated by the following example.

Pump data for Alternative I becomes:

Q = 1, H = 0.5 + 0.5 = 1, P = 1

Pump data for Alternative II becomes:

Q1 = 1, H1 = 0.5 + 0.1 = 0.6, P1 = 0.6

Q2 = 0.5, H2 = 0.5 - 0.1 = 0.4, P2 = 0.20

The total installed pump power P = 0.80

The smaller power requirement for Alternative II is explained by the fact that the surplus pressure in pipe BC is eliminated. In alternative I this excess pressure must be throttled to prevent tapping point C from overflowing and reducing the flow at D.

A better illustration of the pump's operating situation is obtained from drawing the pump and system curves. In Figure 5.29 the respective pump and system curves are shown both separately and in relation to each other. The curves are compiled in accordance with Section 5.1.3. The points marked indicate the situation at the specified flow condition.

The H-Q diagram shows, for example, that pressure boosting is not required at all for total flows Q << 0.46, even if the complete flow is tapped off at D. The largest flow which can pass through the pressure boosting pump is Q ≅ 0.67. This occurs when C is completely closed and is sized for the pressure boosting pump motor. The largest flow at C, D completely closed, is ≅ 0.87. For Q > 0.8, H_B < 0.25, then D is receives no flow unless the pressure boosting pump is started.

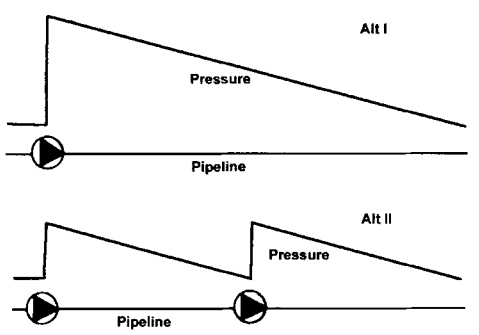
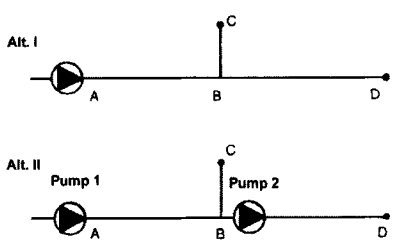


Figure 5.27 Schematic pressure distribution in a pipeline



Data supplied		
Length	Flow	H_{syst}
AB	1	0.5
BC	0.5	0.1
BD	0.5	0.5

Figure 5.28 Example of conditions for pump sizing in a branched pipe system

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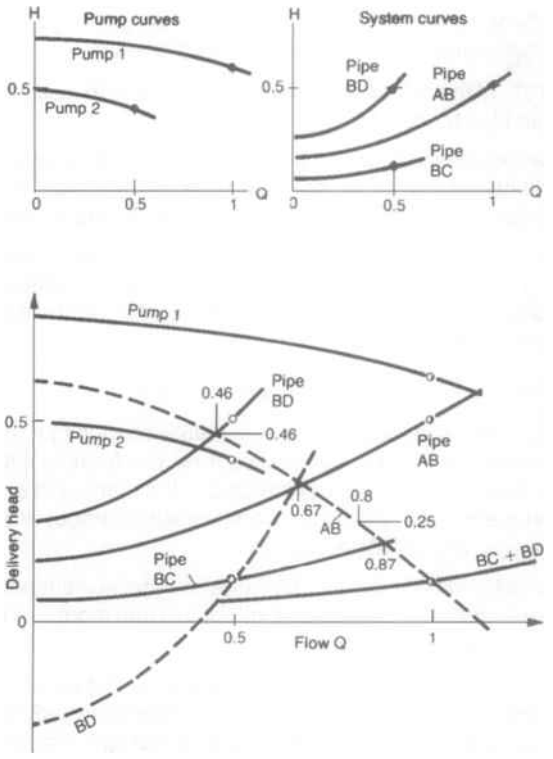

Figure 5.29 Operating conditions for pressure boosting example in Figure 5.28

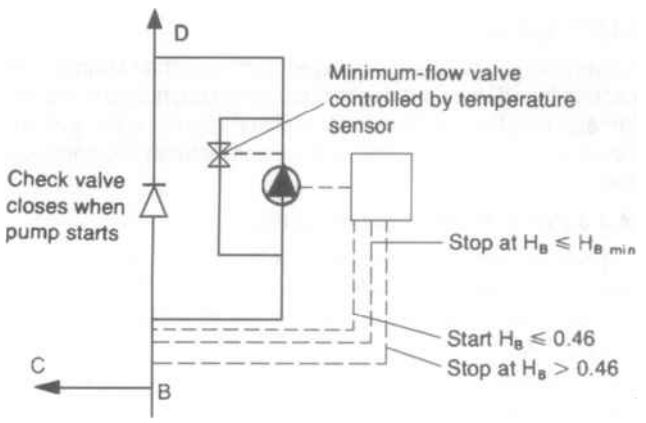

Figure 5.30 Installation of pressure boosting pump

The pressure boosting point can be controlled in the following way and is illustrated in Figure 5.30. To guarantee full flow at D, $Q_D = 0.5$, independent of consumption at C, the pressure boosting pump is started by a signal from a pressure sensor at the branch point when H_B falls below 0.46 and is stopped when H_B exceeds 0.46 by a certain margin. Within the range $0.19 \leq HB \leq 0.46$, the total flow $0.46 \leq Q \leq 0.87$, a situation where flow $Q_D = 0$ can occur.

The input power is transmitted to the liquid, the temperature rises and the risk of seal leakage and damage due to cavitation increases. Excessive temperatures can be avoided by means of a temperature sensor which transmits a signal to a valve which opens a by-pass line. To protect the pressure boosting pump against cavitation in the event of low pressure on the inlet side, which could occur if Pump 1 became damaged for example, a pressure sensor is installed which stops the pressure boosting pump when HB falls below the required cavitation limit, H_{Bmin}.

The most energy conserving method of regulation would be to control the speed of the pressure boosting pump according to the constant pressure criteria in D.

5.3.4 Pressure maintenance

Pressure maintenance, as the name implies, is intended to sustain a certain pressure in a pump installation or circulation system. The liquid being pumped is often hot and, because of the high temperature, has high vapour pressure. The pressure maintenance prevents cavitation at high points in the pipework and pumps, in other words cavitation is avoided. Pressure maintenance also compensates for unintentional leakages and makes up variations in the volume of liquid and pipework due to changes in temperature and pressure.

Typical cases, where pressure maintenance is necessary, are district heating networks and hot water circulation systems in buildings. Pressure maintenance is carried out at a point in the pipe network where it will be most effective, which will vary from case to case. If the pressure maintenance pump is placed on the suction side of the main pump then a constant inlet pressure is maintained, whilst the outlet pressure varies with change in flow and vice-versa.

The simplest method of providing pressure maintenance is by means of a high-level tank. A higher pressure can be provided if required by the introduction of a pressure maintenance pump, Figure 5.31.

The by-pass line ensures a minimum flow to the pressure maintenance pump, even though no liquid is supplied to the main pipeline, thus preventing unacceptable temperature rises in the pump. The by-pass line can also be used to maintain constant pressure for varying flows to the main pipeline.

The main line pressure is equal to the pressure increase due to the pump plus the supply pressure. Constant throttling in the by-pass line causes the pressure level in the main line to vary by an amount ΔH dependent upon the flow in the main line. Vari-

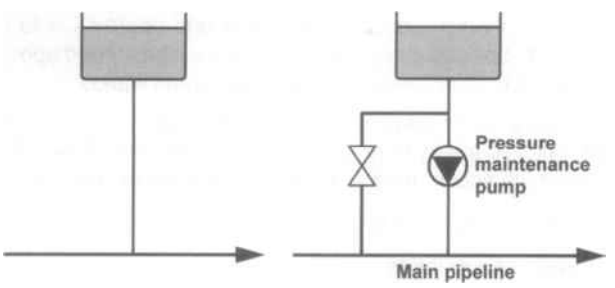

Figure 5.31 Arrangements for pressure maintenance

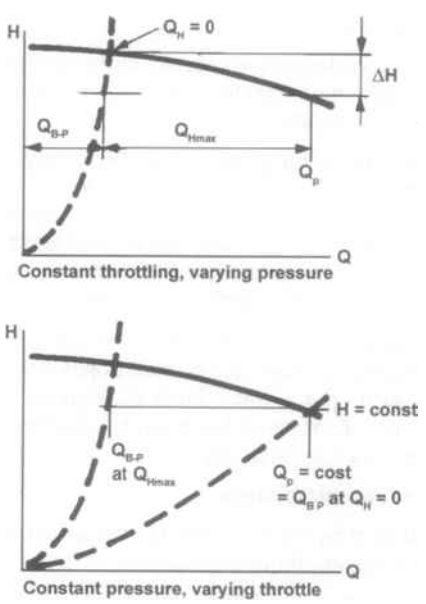

Fig. 5.32 Pressure maintenance pumps for different control methods

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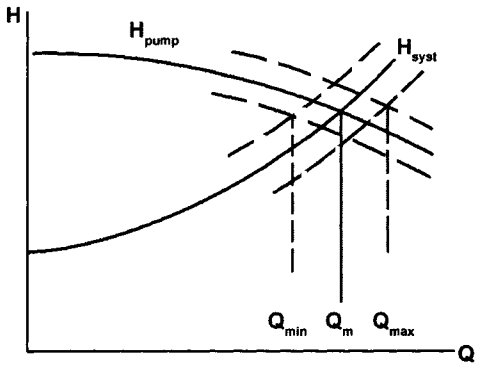

Figure 5.33 Illustration of hydraulic tolerances
——— calculated value — — — — tolerance limits

able throttling enables constant pressure to be maintained, Figure 5.32.

Regulation of pressure maintenance can also be carried out in other ways, for example, by throttling the delivery pipe or by regulating pump speed.

5.4 Pump hydraulic data

5.4.1 Hydraulic tolerances

5.4.1.1 General

All components in the pump installation have tolerances in respect of hydraulic performance. To ensure achievement of a definite flow through the installation it is necessary to consider the hydraulic tolerances when determining pump data, Figure 5.33.

Pump hydraulic tolerances are carefully set out in the pump testing standards. The maximum permissible deviations from the guarantee data are thus defined by specifying the Standard according to which acceptance testing and approval is to be carried out. Special conditions must however be agreed upon in the case of high viscosity and non-Newtonian liquids.

It is however, considerably more difficult to determine the tolerance limits of the system curve, H_{syst}. Examples of the tolerances which accumulate in the complete system curve are:

- Tolerances for pipe diameter
- Pipe bore roughness
- Valves, bends and fittings
- Flow meters and instrumentation
- Variable filter Δp

together with uncertainties in respect of the characteristics of the process liquid at various temperatures and pressures, and the methods of calculation.

Another type of uncertainty, when specifying the pump data, is the future flow and head requirement. This is however, a question which should be clarified before selecting a pump. Rotodynamic pumps can be supplied so that additional impellers can be fitted. Reciprocating pumps can be built to accommodate various sizes of pistons or plungers.

Because of the difficulty in determining the system curve tolerance limits a general allowance is often applied when calculating the necessary pump data. Such allowances are a safety margin and provisions must be made to accommodate them when they are not used usefully.

5.4.1.2 Rotodynamic pumps

The magnitude of hydraulic safety factors which are normally applied to the system requirements are:

- +5% of the calculated differential head due to losses other than H_{stat} for pumps, P > 30 kW
- +2% of the required flow to allow for leakage (may require a larger margin for large complex installations)
- +10% of the calculated differential head due to losses other than H_{stat} for pumps, P < 30 kW

The percentage values given apply to Newtonian liquids. The uncertainty increases in the case of non-Newtonian liquids. The consequences of the allowances are illustrated as follows:

For a mass-produced centrifugal pump, class C, purchased according to a catalogue curve, the tolerances for differential head are ± 6%. For a tolerance of ± 8% for H_{syst} at the desired flow, the allowance will be:

$$\sqrt{0.06^2 + 0.08^2} = 0.10 \text{ i.e. given as } 10\%$$

The calculation means that there is a 95% chance that the actual flow will be equal to the desired flow. The formula also assumes that the production tolerances of the pump and system are independent of each other and equally distributed about their respective calculated values.

The smaller allowance, 5%, for larger pumps is influenced by the stricter delivery tolerance requirements and more extensive system curve calculation.

Another way of approaching the uncertainty of pump/system matching is to specify two duty points. A rated duty and a nominal duty. The rated duty is the flow plus a leakage allowance at the head required to cover all the highest losses. The pump characteristic must pass through this point. The nominal duty is the basic design flow without any leakage at the head required for a new clean system. This point must be on or below the pump characteristic.

Pumps built specifically for a project can have their characteristics "fine tuned" to suit. Impellers can be machined to a specific diameter and the vanes can be modified slightly in the eye and at the tips to change the shape of the characteristic to some degree.

5.4.1.3 Positive displacement pumps

For positive displacement pumps, which have very steep H-Q curves, the system curve tolerances have little effect on the actual flow. The pump flow tolerances are completely definitive, Figure 5.34.

The required flow will most certainly be achieved if the following rules are applied:

- Specify the pump flow as -0% +3% of the design flow.
- Specify the pump suction pressure as the system minimum.
- Specify the pump discharge pressure as the system maximum.

Allowances must be made in the discharge system for overpressure when the relief device operates.

The three rules may not work effectively if the liquid properties, viscosity and compressibility, and the differential pressure vary

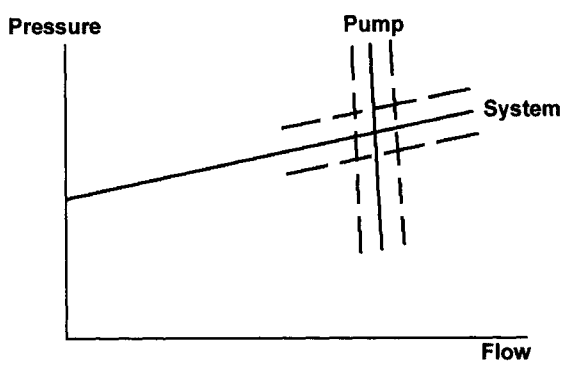

Figure 5.34 Illustration of hydraulic tolerances for positive displacement pumps

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over a wide range. The pump volumetric efficiency could change dramatically resulting in a significant variation in flow. Liquids such as lubricating oils, when the operating temperature range is wide, and liquefied gases, are likely to create problems. Failure to specify the full working pressure range, suction or discharge, could result in the incorrect selection of ancillary products such as pulsation dampers.

To operate any pump at the best efficiency, whether or not the system characteristic is difficult to predict accurately, select a pump which can be adjusted, on site, when the actual equilibrium point can be measured during initial operation. Impeller diameters can be reduced, piston/plungers increased or reduced, or the pump speed can be adjusted. Vee-belts and timing belts are very useful for implementing small changes in speed; the smallest pulley can be specially manufactured to give the desired ratio. The energy cost saving in a single year may easily pay for the extra components. Oversized pumps waste a lot of energy.

5.4.2 Pump H and Q at maximum efficiency η_{max}

On the basis of the installation liquid requirements the mean flow Q_m and maximum flow Q_{max} are determined. Then:

$$H_{Qmax} = H_{stat} + h_{fpipe} + h_v + h_{tol} \qquad \text{Equ 5.16}$$

where:

H_{Qmax} = the pump head at Q_{max} (m)

H_{stat} = static head (m)

h_{fpipe} = head losses in the pipe system excluding the control valve at Q_{max} (m)

h_v = pressure drop across control valve at Q_{max} (m)

h_{tol} = allowance for hydraulic tolerances at Q_{max} (m)

In this way a point Q_{max}/HQ_{max} is established on the intended pump H-Q curve, see Figure 5.35. Except in the case of on-off control systems however the pump will very rarely, if ever, be required to operate at this point. The pump operating point, design point, η_{max}, should preferably occur at a lower flow, as near to Q_m as possible. If the difference between Q_{max} and Q_m is great, more than 15%, and if Q_m is chosen to coincide with the pump's η_{max} then there is risk of over loading the pump at Q_{max}. Normally, however, a flow which is 20% greater than the design flow is not considered to present any problem for rotodynamic pumps providing the manufacturer is consulted and power is available.

Using the above, the following rules of thumb are given for the determining of pump hydraulic specification data for continuously regulated pump installations.

- Determine Q_m and Q_{max} according to the process requirements.
- Calculate H_{Qmax}.
- Select a pump which can cope with Q_{max}, H_{Qmax}. The pump's η_{max} should lie as near to Q_m as possible.
- Check the risk of cavitation, $NPSHr_{Qmax}$, and the required shaft power at Q_{max}. Check with the pump manufacturer if there are any other reasons which may prevent operation at Q_{max}.

The matching of pump flow to Q_{max} when the production tolerances are known can be accomplished by:

- Remachining the pump impeller(s)
- Constant throttling or by-pass
- Speed matching (belt or gear drive)

5.4.3 Minimum pump flow

There are many reasons why it is necessary to ensure a minimum flow for a rotodynamic pump:

- Heating of the pumped liquid.
- Increased cavitation risk when Q approaches zero.
- Increased vibration levels when Q approaches zero.
- Axial thrust and radial force on the pump impeller are greatest at Q = 0.
- The power required for an axial pumps is greatest at Q = 0.

Pump efficiency is low in the vicinity of zero discharge, Q = 0. Most of the power is converted to heat, which must be carried away by the pumped liquid, Figure 5.36. The increase in temperature of the pumped liquid can be calculated from the relationship:

$$\Delta T = \frac{g \cdot H}{c_{sp}} \left(\frac{1}{h} - 1\right) = \frac{P(1-h)}{\rho \cdot c_{sp} \cdot Q}$$

$$\cong \frac{P_{Q=0}}{\rho \cdot c_{sp} \cdot Q} \qquad \text{Equ 5.17}$$

where:

H = differential head (m)

g = acceleration due to gravity (m/s²)

c_{sp} = specific heat (J/kg °K)

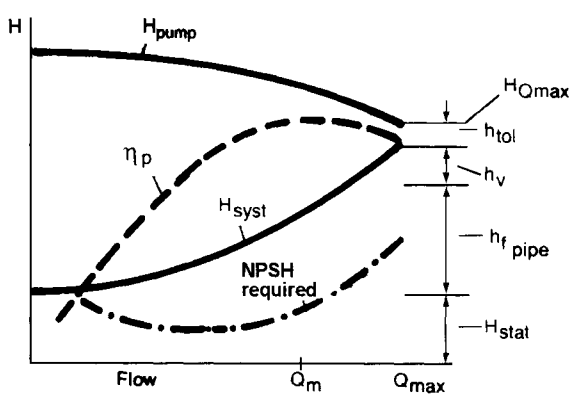

Figure Illustration of recommended pump data

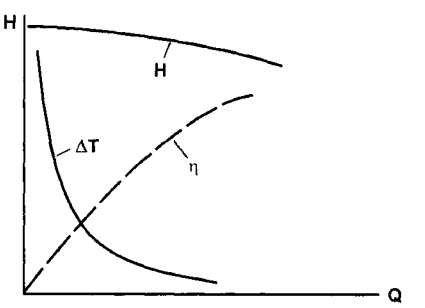

Figure 5.36 Temperature rise of liquid when pumping near to the point of zero discharge

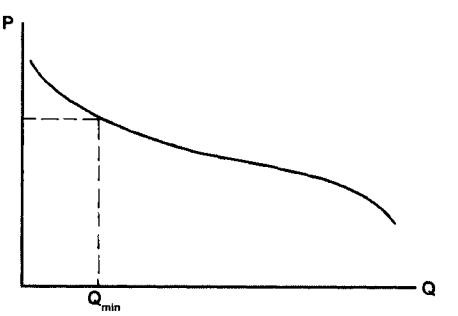

Figure 5.37 Power curve for axial pump

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η = pump efficiency

Q = flow through pump (m^3/s)

ρ = density of liquid (kg/m^3)

P = supplied shaft power (W)

ΔT increases rapidly as the flow through the pump approaches zero ($Q \rightarrow 0$, $\eta \rightarrow 0$). Apart from increased material stresses due to temperature rise, the vapour pressure of the liquid also increases with corresponding cavitation risk and the shaft seal liquid film is also in danger, so that a breakdown of sealing function may be imminent.

For axial pumps, propeller pumps and pumps with high specific speed, the required shaft power increases as flow decreases at constant speed, Figure 5.37.

By having a specified minimum flow through the pump a smaller motor can be used without risk of overloading.

5.4.3.1 Devices for obtaining a minimum flow

A minimum pump flow at Q = 0 in the system can be maintained by introducing a by-pass line into the system, (see Figure 5.38).

The return line should return the liquid to the pump sump in order that a large liquid volume should be available if circulation pumping is protracted. The by-pass flow can be controlled in various ways:

- Constant throttling
- Loaded check valve
- Pressure sensor after the pump
- Flow sensor in the main line
- Temperature sensor after the pump

In cases employing constant throttling, such as an orifice plate or locked needle valve, a certain flow will pass through the by-pass pipe even at Q_{max} in the main line and should then be added to Q_{max} when defining the pump size. A loaded non-return valve or pressure sensor requires a stable pump curve whose differential head falls continuously as pump flow increases.

In the case of positive displacement pumps a relief valve or other overpressure protection device must always be provided as protection against blockage in the discharge line. Flow regulation cannot be carried out by throttling in the main line. Regulation of fixed speed pumps is usually by means of a by-pass line. Other alternatives are speed regulation and variable displacement.

NOTE: Speed regulation using variable frequency inverters to supply standard squirrel cage motors is popular due to ease of use, wide choice and efficiency. Variable speed pumps can overcome all system mismatch problems, usually very efficiently. Introducing a variable frequency inverter into an existing fixed speed installation may cause problems. Inverters can produce unexpected electrical side effects. The best solution is to ask the motor supplier to supply an inverter. Then the supplier is responsible for the inverter and the motor with no divided warranty problems resulting.

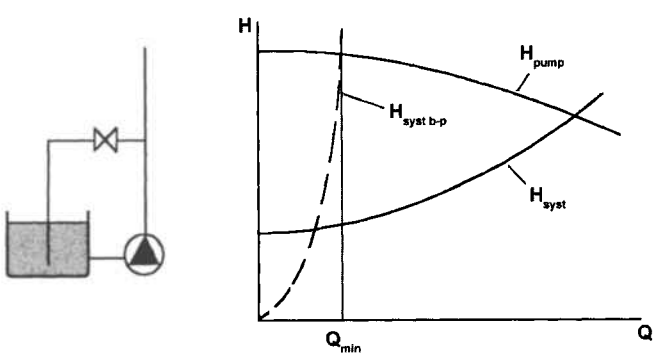

Figure 5.38 By-pass line for a minimum flow

5.5 Water hammer

5.5.1 Hydraulic gradient

In order to obtain a better understanding of the distribution of pressure throughout a pump installation, use is made of the hydraulic gradient. By definition, for a pressure head H_t:

$$H_t = \frac{p^1}{rg} + z \qquad \text{Equ 5.18}$$

where:

H_t = pressure head above liquid surface (m)

p^1 = static pressure above atmospheric (Pa)

ρ = density of the liquid (kg/m^3)

g = acceleration due to gravity (m/s^2)

z = elevation above liquid surface (m)

In this method of representation the pipeline inner positive pressure is the difference between the H_t-line and the elevation of the pipeline, see Figure 5.39. The Figure also shows a high point where there is negative pressure, that is below atmospheric pressure. Parameters which should be observed are the prevailing inner positive and negative pressures as regards sizing for stress and margins of safety against the risk of cavitation. The cavitation risk is greatest in pumps, valves and other installation components having restricted sections together with high points in the system.

The hydraulic gradients for steady flow take on different appearances for different operating points. The extreme value for the H_t-line often occurs, however, in conjunction with load changes associated with a transient, a time dependent process.

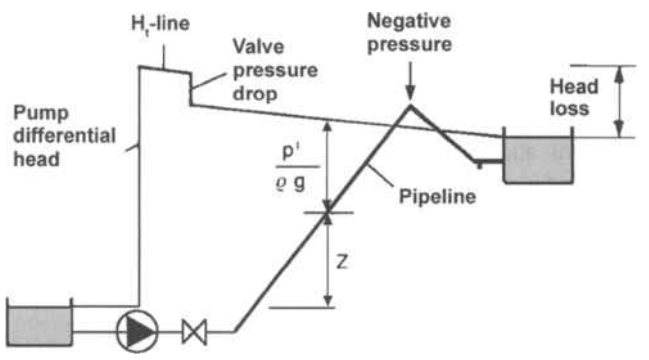

Figure 5.39 Illustration of hydraulic gradient for a pump installation

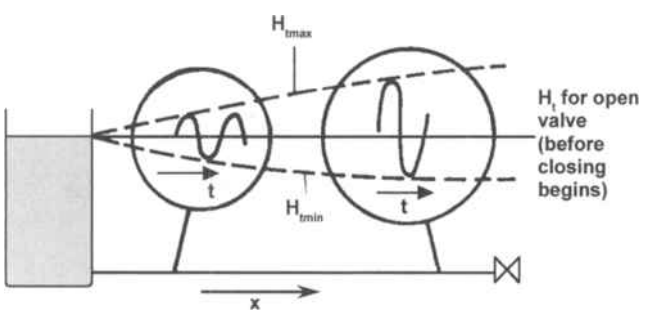

Figure 5.40 Pressure head lines for closing valve

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When closing a valve the pressure increases in front of the valve, see Figure 5.40. Increasing pressure gives rise to a pressure wave, which travels through the pipeline at the speed of wave propagation "a". The magnitude of the wave speed "a" is a function of the liquid and the pipe. Pressure waves would normally travel at the speed of acoustic waves in the liquid, which is a function of the liquid properties only.

$$c = \sqrt{\frac{K}{\rho}} \qquad \text{Equ 5.19}$$

where:

c	=	acoustic velocity (m/s)
K	=	liquid bulk modulus (Pa)
ρ	=	liquid density (kg/m^3)

Water has an acoustic velocity of approximately 1450 m/s. Constraining the liquid within a pipe reduces the acoustic velocity because of the flexibility of the pipe wall, see equation 5.20.

$$a = \sqrt{\frac{g/w}{\frac{1}{K} + \frac{1}{8E}(d_o^4 - d_i^4)}} \qquad \text{Equ 5.20}$$

where:

a	=	wave speed (m/s)
g	=	acceleration due to gravity (m/s^2)
w	=	specific weight (N/m^3)
E	=	Young's Modulus of Elasticity of the pipe (N/m^2)
d_o	=	pipe outside diameter (m)
d_i	=	pipe inside diameter (m)

The elasticity of the pipe wall has a significant effect on the wave speed, but air or gas bubbles have a much more severe effect and can reduce it to 100 m/s. The pressure wave is reflected at the end of the pipe as a negative pressure wave, returns to the valve, is reflected again and so on. In this way a process of pressure oscillation is set up in the pipeline and continues, even after the valve is completely closed, until the oscillations are dampened out by friction. The hydraulic gradient lines H_{tmax} and H_{tmin} connect the highest and lowest pressures which occur at different points along the pipeline during the oscillation process. The greatest pressure variations are to be found in the valve and occur when the valve is closed rapidly. The greatest amplitude of pressure surge is:

$$\Delta p_{max} = \rho \cdot a \cdot v \qquad \text{Equ 5.21}$$

where:

Δp_{max}	=	max. pressure variation (Pa)
ρ	=	density of the liquid (kg/m^3)
a	=	speed of wave propagation (m/s)
v	=	velocity of flow before valve begins to close (m/s)

For water in steel pipes $\rho \cong 1000$ kg/m^3, $a \cong 1100$ m/s, and a flow velocity of 1 m/s creates, for rapid valve closing, a water hammer pressure of 1.1 × 106 Pa = 11 bar ≅ 110 m. The pressure process at the valve cannot be affected before the pressure wave has been reflected from the end of the pipe and returned to the valve. This time is referred to as the pipeline reflection time and is designated t_r.

$$t_r = 2 \cdot \frac{L}{a} \qquad \text{Equ 5.22}$$

where:

t_r	=	pipeline reflection time (s)
L	=	the pipe length (m)
a	=	wave speed (m/s)

If for example the length of pipe is 1100 m, then $t_r \cong 2$ s. All valve closing times ≤ 2 s thus give maximum water hammer pressure.

For a plastic pipe $\Delta p_{max} = 3 \times 10^5$ Pa ≅ 30 m is obtained for valve closing times ≤ 7.36 s. Increasing the closing time reduces the magnitude of water hammer and very slow closing valves result in no line shock at all.

5.5.2 Causes of water hammer

Water hammer in pump installations is caused by changes in flow and especially by rapid flow changes. Rapid flow changes occur during:

- Rapid valve movement
- Starting and stopping of pumps
- Special situations

Since the shock of water hammer can reach considerable magnitude there is risk of damage to the pump installation. Apart from the pressure oscillations which occur with normal variations, the prediction of unintentional operational conditions must also be considered. Accidental pump stoppage caused by a power failure or power cut is one such situation which a pump installation must contend with sooner or later. An estimation of the magnitude of the line shock must therefore be carried out for every installation.

Water hammer caused by cavitation in the pipeline is difficult to estimate. If the absolute pressure somewhere in the pipeline is allowed to fall below the vapour pressure of the liquid, vapour is formed in the pipeline. The original column of liquid divides itself into two which, with different speeds and separated by a growing vapour bubble, flows on through the pipeline. Eventually the two columns of liquid begin to approach each other causing the vapour bubble to implode as the liquid columns reunite, resulting in a large pressure surge. The magnitude of the water hammer shock depends upon the instantaneous difference in velocity of the two columns of liquid at the moment when they reunite, which, in the worst case, can be greater than the steady flow velocity in the pipeline.

Cavitation in pipelines must, therefore, be avoided which also underlines the need to control H_{tmin}. The risk of unacceptable negative pressures are greatest at high points, downstream of valves and pumps and in long suction lines.

The following situations are especially susceptible to water hammer:

- Cavitation due to heating or insufficient cooling in thermal installations
- Trapped air-pockets; pump starts against closed or nearly closed valves in an insufficiently vented system
- Automatic re-start of pump, which rotates backwards after a power failure or after reverse flushing
- Acoustic resonance effects due to periodical disturbances caused by valve oscillations, pump vane passing frequency, positive displacement pump flow variations

5.5.3 Pump behaviour after power loss

5.5.3.1 Pump curves

The pump is sometimes compelled to operate at abnormal operating points especially in the case of pump installations without a non-return valve. Multiple rotodynamic pump installations with parallel operation always have non-return valves fitted to prevent reverse flow back to suction through any stationary

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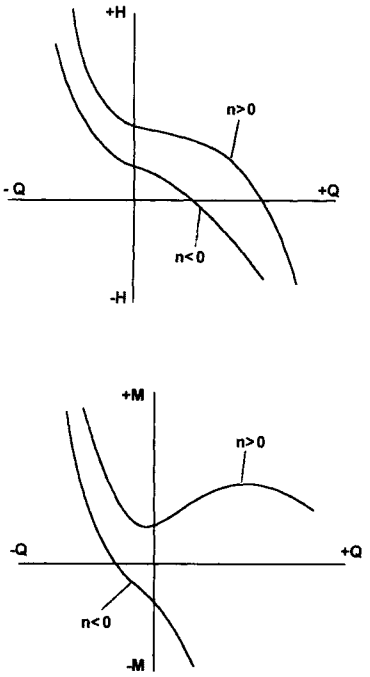

Figure 5.41 Complete performance curves for a centrifugal pump (H = differential head, Q = flow, n = speed, M = shaft torque)

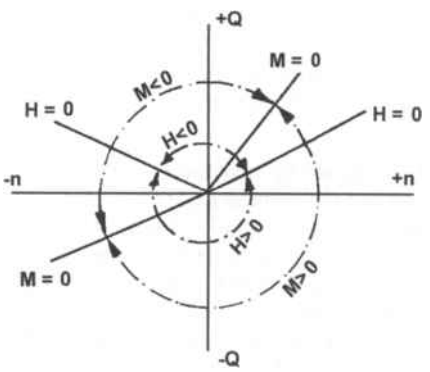

Figure 5.42 Possible centrifugal pump operating conditions

pumps. Single rotodynamic pump installations must have a non-return valve fitted when there is positive static head in the discharge system. Reciprocating pumps do not usually need non-return valves unless a by-pass is used for starting.

Other types of positive displacement pumps may need non-return valves, so consult the pump manufacturer when system characteristics are known. Reverse flow can occur in the event of power failure for example. Inasmuch as reversal of flow or direction of rotation are occurrences which must be considered, a knowledge of pump curve behaviour under these conditions is advisable.

Figure 5.41 shows that a normal centrifugal pump possesses limited pumping capacity even when rotating in the reverse direction and that it functions exceptionally well as a turbine. Values of negative Q indicate reverse flow through the pump. Values of speed, n < 0, indicate reverse rotation of the pump. Reverse rotation can be incompatible with shaft driven oil pumps resulting in considerable damage to mechanical seals and bearings; overspeed may also damage the pump. Pumps with low specific speed tend not to overspeed; overspeeding can be a serious problem with high specific speed pumps.

An important consideration from the point of view of water hammer is the rapidly increasing pressure differential across the pump which occurs with reverse flow and pump speed. Another way of illustrating a possible pump operating condition is shown in Figure 5.42.

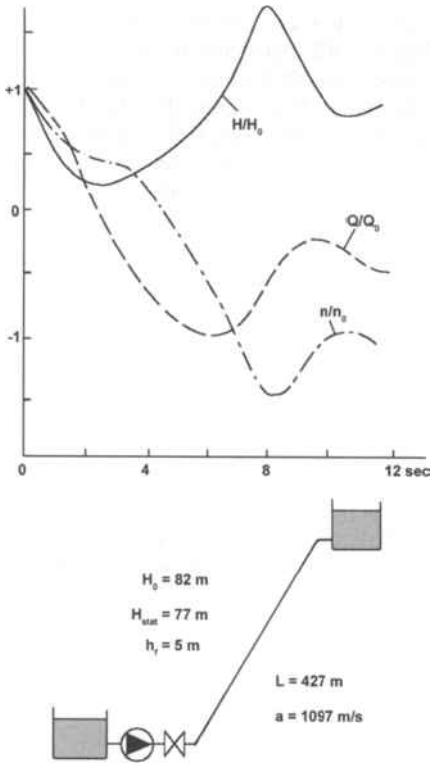

Figure 5.43 Transient process during power failure conditions at pump

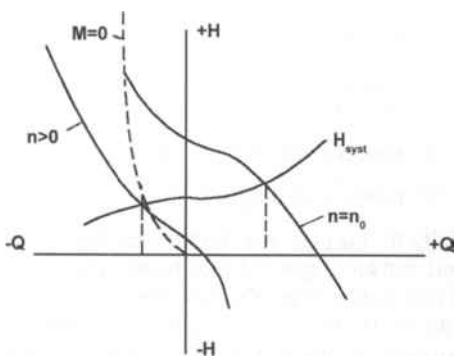

Figure 5.44 Steady, initial and final conditions during power failure

5.5.3.2 Pump stops without non-return valve

The transient process in the event of power failure is illustrated by an extreme example, see Figures 5.43 and 5.44. The installation is equipped with a slow-closing valve which remains open during the process.

When the driving torque fails, the impeller is braked by the hydraulic torque. The decreasing speed also reduces Q and H for the pump. The flow through the pump changes direction whilst the speed is still positive. The pressure in the pump reaches its maximum value. The direction of rotation also changes a few seconds later. Very high pressures occur in the pump when retarding reversed flow at high reversed speed.

The pressure oscillations are eventually dampened out and a steady flow condition is obtained. The pump reaches its so-called runaway speed.

5.5.4 Protection against water hammer

5.5.4.1 Protection equipment

Protection equipment is necessary when the pump installation would be damaged by the resultant:

- Excess pressure
- Risk of cavitation
- Negative pressure in the line

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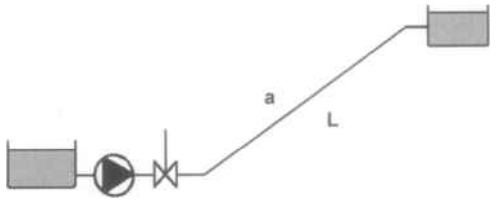

Figure 5.45 Pump installation with slow-closing valve

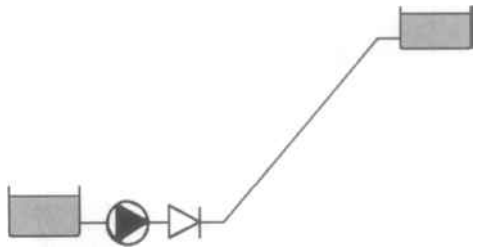

Figure 5.46 Pump installation with non-return valve

- Reverse running and overspeed

Rotodynamic pumps should usually be fitted with non-return valves; this is not considered protection in this context. Installations subjected to repeated water hammer must be designed to withstand fatigue.

Detailed water hammer calculations are complicated and costly. Specialist consultants can be employed to investigate the effects and make recommendations for protection. This raises the question as to when a detailed investigation is necessary. For certain installations simple estimations can give an indication as to the magnitude of water hammer.

5.5.4.2 Valve closing time T

In Figure 5.45 the pump starts and stops against a closed valve. The pressure changes at the valve in connection with closing can be approximated by:

$$\pm \Delta H = \frac{2 \cdot L \cdot v}{g \cdot T} \quad \text{for } T > 2 \cdot \frac{L}{a} >$$

$$\pm \Delta H = \frac{a \cdot v}{g} \quad \text{for } T \leq 2 \frac{L}{a} \qquad \text{Equ 5.23}$$

where:

L	=	pipe length (m)
v	=	initial flow velocity (m/s)
g	=	acceleration due to gravity (m/s^2)
T	=	valve closing time (s)
a	=	wave speed (m/s)

The above formula assumes a linear valve closing characteristic, i.e. linear reduction in flow in relation to time.

5.5.4.3 Pump stops with non-return valve

In Figure 5.46 the pressure changes with non-return valves in conjunction with the pump stopping are approximated by:

$$\pm \Delta H = \frac{2}{\frac{g \cdot C}{v \cdot L} + \frac{K}{H}} \qquad \text{Equ 5.24}$$

Equation 5.24 does not apply to special non-return valves which have delayed action and/or anti-slam features.

Calculation example:

The following data applies to a pump installation with a non-return valve; pipe length L = 1700 m, delivery head H = 22 m and flow velocity v = 1.3 m/s in the pipeline. Estimate the magnitude of pressure surge when the pump stops.

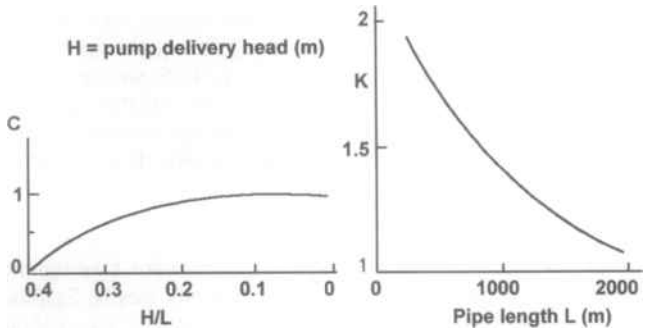

Figure 5.47 Values for C and K in Equ 5.24

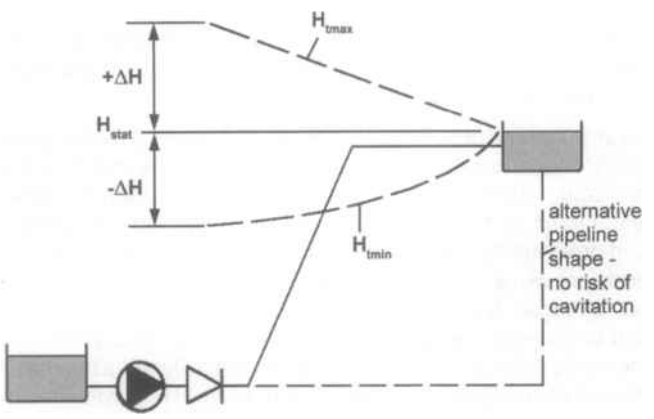

Figure 5.48 Reviewing hydraulic gradients

$$L = 1700 \text{ m} \rightarrow K = 1.1,\ H/L = 22/1700 \rightarrow C = 1$$

Then according to equation 5.24

$$\pm \Delta H = \frac{2}{\frac{g \cdot 1}{1.3 \cdot 1700} + \frac{1.1}{22}} = 37 \text{ m}$$

5.5.4.4 Pump stops without non-return valve

The installation is the same as in Figure 5.45. The pump stops because of power failure. The slow-closing valve remains open.

$$\text{If } \frac{h_f}{H} > 0.2 \qquad \text{Equ 5.25}$$

then

$$H_{tmax} \leq H$$

where:

h_f	=	pipe system head loss (m)
H	=	pump delivery head (m)

The size of the pressure pulse, -ΔH, is estimated from Equ 5.24 using the values from figure 5.47.

5.5.4.5 Hydraulic gradients

The rules of thumb take no account of the shape of the hydraulic gradient along the pipeline. The gradients shown in Figure 5.48 are based on estimates derived from experience. It is, however, possible to conclude that there are definite risks of cavitation in the case of the pipeline shape as represented by the continuous line. The chain-dotted line routing eliminates this risk. It then remains to be shown whether or not H_{tmax} at the valve is acceptable.

By means of these simple estimations it is possible to judge many non-critical cases, usually having short pipe lengths and low flow velocities. In cases of doubt it is advisable, as the next stage of investigation, to refer to the sizing diagrams which are published in technical literature etc. These cover more thor-

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ough calculations for pump stops without non-return valves for certain simple pump installations and also quote figures for H_{tmax} and H_{tmin} at mid-point in the pipeline. (See work by Kinno and Kennedy.) If there is still cause for doubt then the only solution is to carry out a detailed digital computer simulation using the Method of Characteristics. See also work by Parmakian, or an analogue computer simulation.

5.5.4.6 Protection equipment

Water hammer is avoided during normal operation by ensuring that load changes are carried out sufficiently slowly. Starting and stopping of pumps is carried out against closed valves and/or by using by-passes. Extra protection equipment is therefore usually installed in order to protect the pump installation against power failure.

A large proportion of protection equipment attempts to eliminate the basic cause of water hammer, namely the too rapid flow reduction resulting from a power failure.

A surge column of sufficient diameter and height provides good protection, see Figure 5.49. In the event of power failure the non-return valve closes and isolates the pump from the pipeline. Liquid from the surge column flows out into the pipeline and prolongs the retardation process, weakening the negative pressure wave. The column refills when the flow in the pipeline has changed direction. The mouth of the column can be throttled to increase dampening in the system. A disadvantage of the surge column is that it must be at least as high as the maximum pressure head at the non-return valve. This problem can be avoided by means of an air chamber, Figure 5.50.

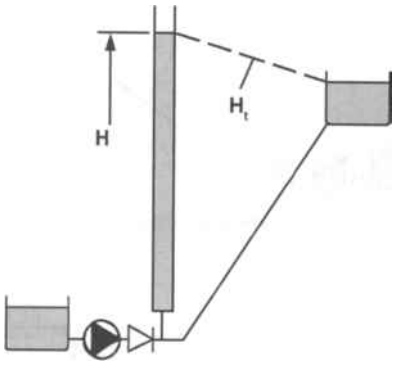

Figure 5.49 Surge column

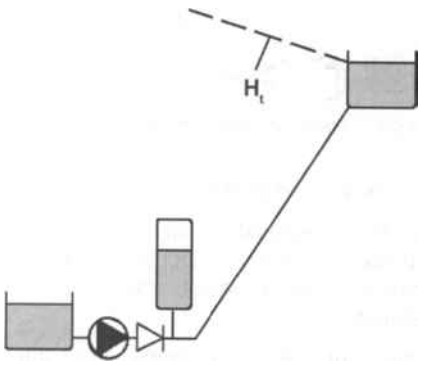

Figure 5.50 Air chamber

During normal operation the air pressure in the air chamber is equal to the pump delivery head. When pumping stops liquid flows out of the air chamber and the air pressure reduces. The greater the volume of air the smoother the process. The system can be suitably dampened by throttling the chamber at the flow inlet. A disadvantage with the air chamber is the need to provide a compressor to replace the air which is dissolved in the liquid and carried away in the pipeline. This may still be considerably cheaper than building a much taller water column.

The problem of air dissolution in the water can be greatly reduced by using a proprietary surge arrester which uses a nitrogen charge stored within an elastomer envelope. Direct contact of gas and water is eliminated. The elastomer envelope is not 100% gas tight and a very small gas loss is experienced. The gas volume can be topped up as necessary during routine maintenance.

The retardation of the pump rotor is also affected by the moment of inertia of the rotating parts. By fitting the pump with a flywheel (Figure 5.51) it is possible to prolong the run-down time. The method is most effective for high speed pumps and moderate pipe sizes. A disadvantage is increased starting difficulties. The starting problems can be alleviated by using modern soft starters. Variable speed pumps should not have starting problems.

With low or negative static heads there is a risk of negative pressure on the pressure side of the pump. The pressure drop across the pump is reduced by means of a by-pass line to augment forward flow when the pump has stopped, Figure 5.52.

The other main group of protection equipment is intended to eliminate water hammer at a somewhat later stage of the transient process.

Negative pressure, with the associated cavitation risks can be avoided by placing a surge tank at critical points (Figure 5.53). In the event of negative pressure, liquid flows from the tank into the pipeline. The tank is refilled during normal operation via a separate level controlled filling line.

An alternative method is to let air into the pipeline when the pressure falls below atmospheric pressure. The pipeline must be vented, however, prior to re-starting. In installations having large static delivery heads it is often the excess pressures which occur upon reversal of the liquid column and in bringing it to rest, which are most critical. These pressures can be reduced with the aid of control valves, (Figures 5.54 to 5.56).

In order not to amplify the initial pressure wave, the negative pressure wave, the controlled by-pass valve remains closed, in principle, until the flow at the pump is reversed. It then opens

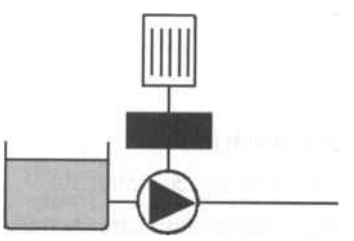

Figure 5.51 Pump with flywheel

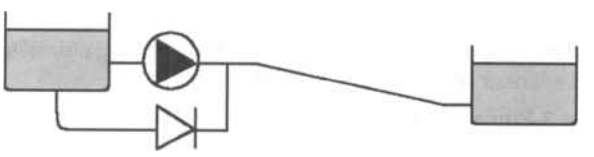

Figure 5.52 Pump with reverse by-pass

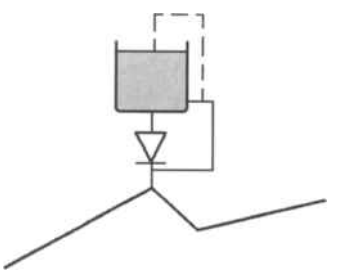

Figure 5.53 Surge tank

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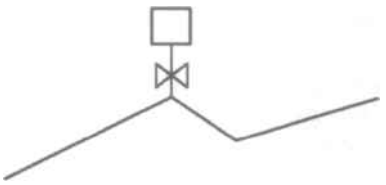

Figure 5.54 Automatic air bleed valve

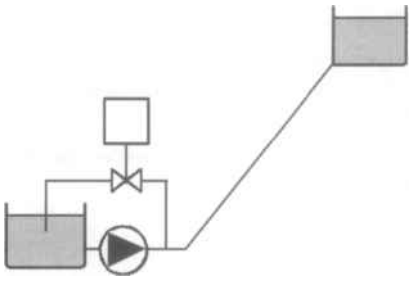

Figure 5.55 Controlled valve in by-pass line

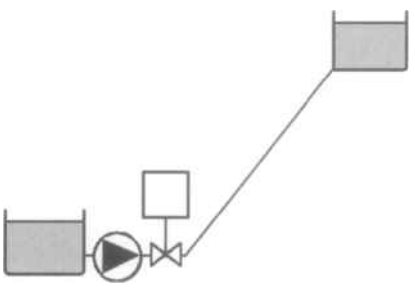

Figure 5.56 Controlled valve in main line

rapidly, thereby relieving pressure in the pipeline and finally it closes slowly.

The controlled valve in the main line should remain open until the flow at the pump reverses. It then closes rapidly to its optimum position, ideally to retard the reversed flow with optimum distribution of pressure drop between the pump and valve.

Pump reverse rotation can be prevented by means of a reverse running clutch, Figure 5.57. The reversed pump flow is then retarded more smoothly and the pressure surge is dampened. The risk of reverse overspeed is eliminated.

Apart from the pure pressure surge dampening characteristics, the choice of protection equipment is also affected by other factors; for example, liquid characteristics, construction, maintenance and procurement costs. Each case must be judged on its own merits as to the most suitable choice.

5.6 Pressure pulsations

As shown in Section 5.5, pressure surges can occur due to isolated operating conditions such as start-up, shut-down and valve operations. These are not the only conditions under which rapid pressure changes occur. Some systems run continuously with pressure pulsations.

5.6.1 Rotodynamic pumps

It is generally thought, and often stated in literature, that rotodynamic pumps produce a continuous steady flow pattern without pressure pulsations. Axial flow pumps probably come closest of all rotodynamic pumps to fulfilling this requirement; mainly because of the low differential head produced.

Mixed flow pumps tend to be used for low differential head applications with high flow rates and consequently have wide impeller tip widths. Wide impeller passages, in the axial direction, produce uneven velocity streams with the liquid. The liquid leaving the pump discharge connection is not a steady stream but a mixture of several flow streams. As the liquid flow pattern stabilises in the discharge pipework, pressure pulsations and

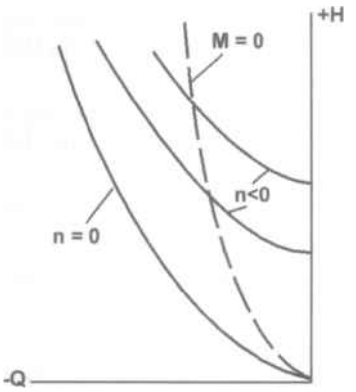

Figure 5.57 Effects of reverse running clutch

flow variations can be measured. If flow measuring devices are to be used close to the pump discharge some form of flow straightening may be necessary.

Centrifugal pumps operate at a wide range of speeds. As speed is increased the circumferential flow pattern between adjacent vanes is modified. Liquid tends to be forced closer to the advancing vane and dragged away from the receding vane. It has been suggested that the flow between a pair of vanes will segregate into three distinct flow streams moving at different velocities. The action of one or two cutwaters produces additional pulsations. It has been proposed in new test standards that fluctuations in flow readings of ± 3%, smallest tolerances, are acceptable. Obviously flow variations of this magnitude will have associated pressure pulsations.

Centrifugal pumps can have additional problems. When operating at low flows some pumps surge in the suction pipework due to recirculation in the impeller eye. Pump manufacturers should be able to quantify the onset and magnitude of suction surge.

5.6.2 Positive displacement pumps

The flow variations of reciprocating pumps are well documented, even if incorrectly. These flow variations are of sufficient magnitude to produce pressure pulsations which create pipework vibration which do not require special instrumentation to detect. Some other positive displacement pumps produce pressure pulsations. Lobe pumps and gear pumps with few teeth deserve special attention.

Newer test standards recognize the problems caused by pressure pulsations and include test requirements. However, testing on any test rig will not reproduce site conditions or site pulsation levels. Pressure pulsations can be amplified by pipework resonance and cause severe problems. Problems with site pipework can only be detected by computer simulation or site operation.

Most pressure pulsation problems can be controlled by pulsation dampers, see Chapter 11, Section 11.6, or by simple pipework modifications to eliminate resonance. The proprietary gas-charged surge arrester, mentioned in Section 5.5.4.6, can also be used to attenuate pressure pulsations created by continuous cyclic flow variations. Surge arresters can be sized to provide an attenuation up to 90%. Correct sizing is dependent upon a fairly accurate prediction of all operating pressures.

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6.1 Introduction

When dealing with liquid transportation, a pump installation is sized to handle a maximum flow which, in practice, might never occur. This principle of sizing is nevertheless correct, since inadequate pumping capacity on isolated occasions can lead to the most damaging consequences. It is also obvious that sensible ways of varying the flow are important, not only for the functioning of the pump installation itself but also for the process for which the pump is included as a major or minor component.

Plant procurement costs generally amount to a small proportion of the total pump investment cost of a plant. But not always! Some pumps cost over $2M each and the remainder of the installation is just pipework. In most installations the functional quality of the pump may be the critical factor for the overall operation and its associated running costs. Advanced methods of flow control, e.g. speed regulation of pumps, are nowadays the best methods of varying the flow, but with the disadvantage that initial investment costs for the pump equipment are increased. The advantages are reduced running costs overall, smaller requirements for storage tanks and sumps, smaller electric cabling, and a reduction in starting current.

With existing installations, the method of flow regulation affects, most of all, the energy costs. As a result of increasing energy costs, the optimum methods of pump control have changed greatly from those with low initial costs and high energy consumption to those with relatively high investment costs and good power economy. For pumps with a power requirement greater than 10 kW, it will even now be beneficial to convert to more power-saving methods of regulation. See also Chapter 14.

Pump flow can be controlled in steps, discrete flow regulation or smoothly, continuous flow regulation or infinitely variable flow regulation. Combinations of the two methods are also used. The choice of method is determined by the overall requirement of the pump installation, variation of flow, timescale, operating economics, initial cost and technical possibilities.

In this Chapter, pump, may also mean pumps, for example multiple pumps of the same size operated in parallel. When the detailed description of the installation is explained it will become obvious whether a single pump or multiple pumps are involved.

The operating point of the pump at a particular speed is the intersection of the system characteristic and the pump curve as in Figure 6.1. At this point, the energy dissipation of the system is equal to the energy supplied by the pump.

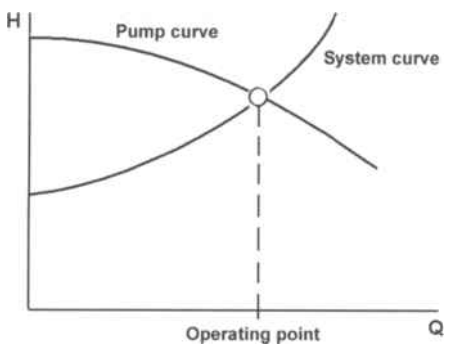

Figure 6.1 The operating point of pump and system

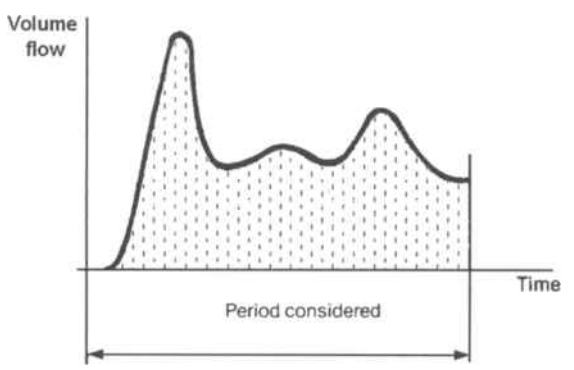

Figure 6.2 Flow demand as a function of time, running curve

6.2 Variable flow requirements

Whether or not a pump needs flow regulation is dependent upon the pump type and the nature of the process involved. The pump flow may not match process demands because of:

- Safety margins and hydraulic tolerances
- Failure to utilise full production capacity
- Variations in requirement caused by climatic variations, e.g. summer, winter, rain, drought
- Variations in production processes

Flow variation with time can be described either by a running curve, Figure 6.2 or by a constancy curve, Figure 6.3.

In the constancy curve, time has been summated for all periods of flow having the same magnitude. A point on the constancy curve gives the time during which the flow at least amounts to a particular value.

Constancy curves can be made up for various periods of time ranging from a full 24 hours to the lifetime of the installation. Constancy curves are usually made up for a period of 1 year.

Unfortunately, constancy curves are not that easy to compile, but a simple analysis of the actual liquid transportation often gives adequate guidance for the determination of maximum, minimum and average flow. The mean flow, for example can be determined from the annual production capacity and maximum flow from peak consumption.

The pump size is obviously determined by the maximum flow figure, whereas pump economy is determined by the duration of flow. Economic assessments are easily made if the volume flow is known at 100%, 50% and 0% constancy. It would be even better, of course, if the values at 25% and 75% are also known.

The following simple equation is obtained by the use of Simpson's Rule for integration:

$$Q_{mean} = \frac{1}{6} \cdot Q_{100} + \frac{2}{3} \cdot Q_{50} + \frac{1}{6} \cdot Q_0 \qquad \text{Equ 6.1}$$

where:

Q_{mean} = mean flow (m^3/s)

Q_{100} = flow at 100% constancy, i.e. the flow taking place 100% of operational time ($Q_{100} = Q_{min}$) (m^3/s)

Q_{50} = flow at 50% constancy (m^3/s)

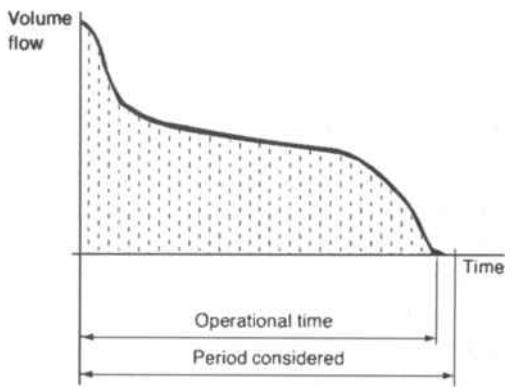

Figure 6.3 Constancy curve for flow

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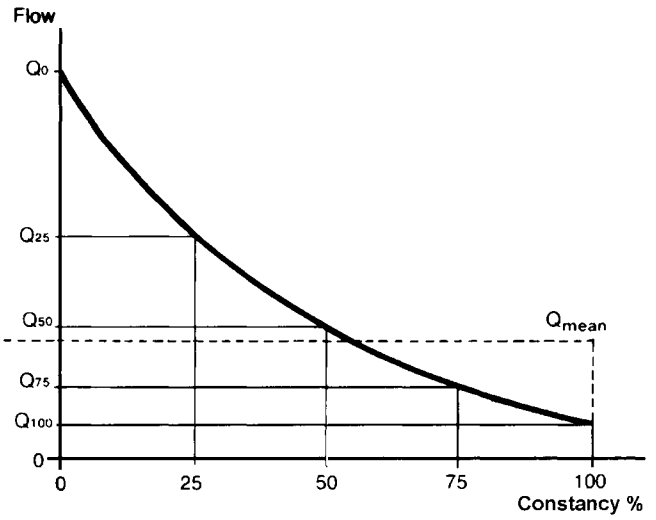

Figure 6.4 Graphical representation of Equations 6.1 and 6.2

Q_0 = flow at 0% constancy ($Q_0 = Q_{max}$) (m^3/s)

or if more points on the curve, see Figure 6.4, are known:

$$Q_{mean} = \frac{1}{12} \cdot Q_{100} + \frac{1}{3} \cdot Q_{75} + \frac{1}{6} \cdot Q_{50} + \frac{1}{3} \cdot Q_{25} \qquad \text{Equ 6.2}$$

where Q_{75} and Q_{25} are expressions for the flow at 75% and 25% constancy.

6.3 Flow regulation

6.3.1 Methods

As Table 6.1 shows, flow can be regulated in many ways. Throttle regulation, by-pass regulation and overflow by-pass regulation all mean that the control takes place outside the actual pump itself. Some of the flow produced by the pump is thrown away, wasting energy.

Method	Advantages	Disadvantages
Discrete variations		
Start-stop control	Cheap, easy to operate	Flow surges, water hammer, wear and tear on switchgear
On-off by-pass return	Relatively cheap, easy to operate	Flow surge problems eliminated with slow acting valve
Pole-changing motors	Relatively cheap, easy to operate	Must stop to change speed, limited number of speed options, flow surges, water hammer
Multi-speed gearbox	Easy to operate	Must stop to change speed, flow surges, water hammer, may need maintenance staff intervention (1)
Vee belts and pulleys	Cheap	Must stop to change speed, flow surges, water hammer, needs maintenance staff intervention
Continuously variable		
Throttling	Cheap, easy to operate	Wastes energy, not for solids handling applications
Modulating by-pass return	Easy to operate	Wastes energy, not for solids handling applications
Overflow by-pass return	Easy to operate	Wastes energy
Variable speed	Energy efficient	Can be costly to install
Adjustable blade angle	Energy efficient	Axial pumps only, low differential head
Adjustable inlet guide vanes	Energy efficient	Limited flow range, can be costly to modify pump
Part loading	Energy efficient	Archimedean screw only
Adjustable stroke	Energy efficient	Reciprocating pumps only, very costly in larger pumps

(1) Portable engine driven pumps frequently use semi-automatic transmissions, no need to stop to change gear.

Table 6.1 Pump flow regulation methods

Other methods change the actual performance of the pump including power consumption, either within the pump unit or by means of variable filling, e.g. Archimedean screw pumps and displacement regulation. As can be seen from Table 6.1, the only universally applicable method is speed regulation, i.e. it can be used for all types of pumps, from the smallest dosing pump with power requirements of 10 to 100 W to the largest possible pump units in power stations with power requirements up to 60 MW.

There are no rules against combining various methods, e.g. the parallel operation of several units with continuous regulation by means of throttling or speed regulation. The choice of optimum methods requires great experience and often extensive economic and process technical analyses. Sometimes there are also special problems to be considered, such as noise levels from control valves when throttling.

As mentioned earlier, it is possible to combine control methods. If a modulating by-pass was fitted to most of the discrete variation methods the problems of surge and water hammer could be eliminated.

Series and parallel operation are dealt with in Chapter 5, Sections 5.3.1 and 5.3.2 respectively. Examples of pump performance for pumps with adjustable impeller blades, or with adjustable inlet guide vanes, pre-rotation, are shown in Figures 4.21 and 4.22 respectively. Archimedes rigid screws flow regulation is dealt with in Section 1.5.3, Chapter 1. Metering pump flow regulation is dealt with in Metering pumps in Section 1.5.19, Chapter 1.

Where changes in flow requirements are needed over longer periods, more than about six months, say, it may be economic to change the diameter of a centrifugal pump impeller or piston/plunger or modify the blade angles on a propeller. See also Chapter 4, Section 4.2.2.

6.3.2 Control parameters

Control parameters may be flow, pressure, or some quantity derived from these such as level, temperature, concentration and so on. Some examples of these are shown in Table 6.2.

Control parameter	Sensor type	Example of application
Discrete level control	Level switch	Fresh water reservoirs Emptying separators
Proportional level control	Level transmitter	Process control on reactors and vessels
Constant flow	Flow meter	Process control for variable pressure systems
Discrete pressure control	Pressure switch	Charging vessels and accumulators. Pressure maintenance
Constant pressure	Pressure transmitter	Process control of vessels and accumulators
Proportion control for mixing	Flow meter in each circuit	Chemical processing Water treatment
Proportion control for mixing	Flow meter in mixer outlet	Chemical processing Water treatment

Table 6.2 Some control parameters

The principles of regulation can also be shown by the H-Q diagram, see Figure 6.5 for the simplest cases.

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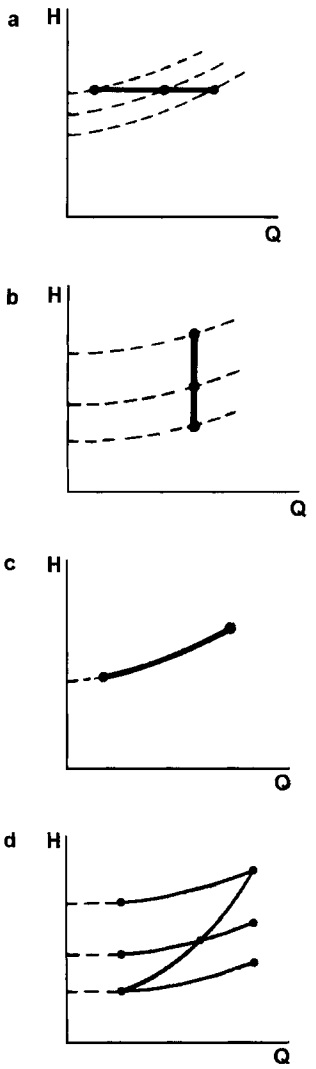

Figure 6.5 Pump regulation for system configurations
(a) constant pressure for variable system
(b) constant flow for variable system
(c) variable flow for a fixed system
(d) variable flow for variable system

6.4 On-off control of constant speed pump

6.4.1 Principles

On-off control means that one or more pumps are started and stopped, or loaded and unloaded, systematically, according to the capacity requirement. Loading and unloading is accomplished by an isolating valve in a by-pass. When the flow requirement does not coincide with the pump capacities the pumps will work intermittently at a starting frequency of 1 to 15 starts per hour for small pumps, reducing to 1 to 4 starts per hour for larger pumps. To account for the difference between instantaneous flow demand and the supply flow, some form of liquid storage or reservoir must be provided.

On-off control and load/unload control is arranged automatically so that pumps are started at a high level in a storage vessel on the suction side of the pump or at low level in a storage vessel on the discharge side. When pumping between a suction tank and a discharge tank there is no need for other vessels or tanks to store the imbalance between supply and demand. The storage capacity will be included in the suction/discharge tanks.

For systems using pressure rather than level control, gas charged storage vessels, called accumulators, are used to store the imbalance. Pumping begins with a low pressure signal from the accumulator and stops at a high pressure signal. The accumulator(s) must be sized so that motors do not start more frequently than their specification allows. A combination of on-off and load-unload can be very effective with larger pumps to eliminate starting problems and also increases the efficiency of load-unload used in isolation.

6.4.2 Costs

Of the discrete flow control methods, the on-off and load-unload methods are the most usual. The on-off control method is the cheapest of all methods to install. The additional costs for full automatic control may be limited to a single level switch, extra control contacts in the motor starters and some wiring. The starters for the pump motors are required for every installation, control contacts are standard with some models.

However, consideration must be given to the following when additions are necessary to the pump installation:

- Cost of extra storage in which to hold the liquid during the regulation cycle
- Additional costs for heavy duty starting equipment since this has to be selected to cope with a very large number of regular starts
- Additional costs for equipment to assist in the alleviation of water hammer when starting/stopping and loading/ unloading
- Additional costs for pipework and fittings if these have to be designed to withstand the fatigue stresses and vibration induced by water hammer

6.4.3 Starting/loading problems

The number of permitted starts per hour is a critical factor in total initial cost for the pump installation. The type of starting used, direct-on-line, DOL, or star-delta, S-D, or a modern soft start system affects the frequency of starting. Small motors can be started 20 times per hour without problems, depending upon the operating conditions. Larger motors, 125 kW and above, have strict restrictions on starting, 4 starts per hour maximum is very common. Experience shows that the number of starts chosen can be 5 to 10 starts per hour. Thus there will be 20,000 to 50,000 starts per year and up to the time of financial write-off, the number of starts will amount to several millions, thus some components have to be sized to withstand fatigue if precautions are not taken.

Even if the starting time of the pump is short, about 0.25 s with DOL and about 1 s to 1.5 s for S-D, this can put a heavy strain on the mains supply.

Loading and unloading by automatic by-pass removes the electrical problems associated with starting and stopping. This approach should be seriously considered as motor sizes increase. A slow acting by-pass valve can eliminate the surge and water hammer problems. Fatigue may have to be considered if parts of the discharge pipework system cycle from low pressure to high pressure with every loading cycle. Pressure system code compliance, without fatigue considerations, is usually limited to 7000 pressure cycles. Much larger cycle counts may reduce the allowable stress levels to 50% resulting in design difficulties or routine replacement of piping sections.

The selection of the by-pass valve is of critical importance for the reliability of the whole system. Water systems can impose unexpected difficulties due to dissolved air and the deposition of salts. By-pass valves must have a throttling capability for the smooth transition from full flow to zero flow.

Reciprocating pumps can avoid the starting and by-pass valve problems by using suction valve unloaders. During periods of zero flow a pneumatic or hydraulic control system lifts the suction valves off their seats. No discharge flow is possible and "slugs" of liquid recycle in the suction manifold. When flow is re-

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quired the suction valves are lowered, in sequence, to commence the normal pumping operation. If gas charged pulsation dampers are fitted they can be slightly oversized to provide some surge protection.

6.4.4 Stopping/unloading problems

A liquid flowing in a lengthy pipeline possesses considerable kinetic energy. When the liquid decelerates with the stopping of the pump, large pressure variations occur. This can be calculated and is, for abrupt stops:

$$\Delta H = \frac{a \cdot v_o}{g} \qquad \text{Equ 6.3}$$

where:

ΔH = change of pressure head (m)

a = wave speed (m/s)

v_o = initial liquid velocity (m/s)

g = gravitational acceleration (m/s^2)

In order to reduce the variation of head the flow has to be decelerated over a period of time, t_o (s), which can be roughly assessed in a pipeline of length L (m) as:

$$t_o = (15 \text{ to } 60) \cdot \frac{L}{a} \qquad \text{Equ 6.4}$$

In the case of pipelines with high points or with low resistance to fatigue, the stopping time may have to be doubled. The normal arrangements used to increase the stopping time or to reduce the stresses are:

- Slow-closing valves
- Air-chambers or pulsation dampers
- Pumps with flywheels
- Surge columns
- Overpressure and underpressure valves

See Chapter 5, Section 5.5.4 for full details.

In all cases the flow continues for a time t_o. To avoid sucking air into the pump and pipeline, a sufficient supply of liquid must be available on the suction side of the pump, whilst that same volume of liquid must not cause trouble on the delivery side, through surge for example.

For air chambers, the volume of liquid is contained in the air chamber and in other cases, the storage tank or pump sump has to supply the corresponding increase in volume. When the flow requirement or inflow is negligible, which is the most unfavourable condition, the mean flow during the stop cycle can be estimated to be about ⅔ of the maximum flow. The partial storage volume V_o (m^3), between stop signal and stop flow will, for the decelerating time t_o s, be as before:

$$V_o = (10 \text{ to } 4) \cdot \frac{L}{a} \cdot Q_p \qquad \text{Equ. 6.5}$$

where:

Q_p = pump flow (m^3/s)

With longer pipelines, especially for installations with slow closing valves, a large part of the operational period may be taken up by the start and stop cycles, see Figure 6.6. During these transients, throttle regulation or modulating by-pass takes place, see also Sections 6.7 and 6.8. Power consumption is determined by the relationship between the times for pure on-off operation and those for valve operation. Throttle regulation and modulating by-pass can increase energy consumption by anything up to 50 %.

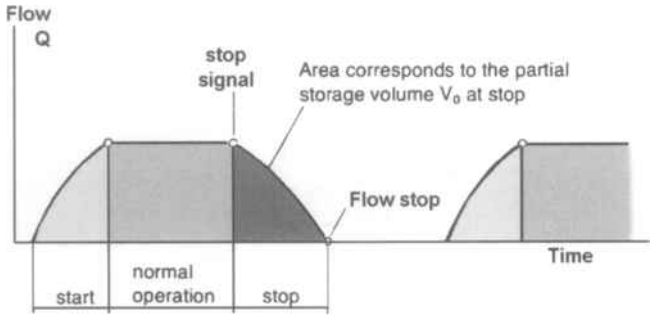

Figure 6.6 Flow variation for on-off and load/unload control with long pipeline

6.4.5 Operational sequences

There is a choice of several different operational sequences for switching in and out parallel operation units. Critical factors to be considered are:

- Number of permitted starts per pump per hour
- Number of permitted starts during the lifetime of the installation
- Permitted variations in flow
- Costs of storage tanks, sumps and accumulators

The differences between various possible sequences, designated A, B and C, are shown in the following three schematic examples, Figures 6.7, 6.8 and 6.9 respectively, applied to an installation with two identical parallel operating pumps P1 and P2. It becomes clearest if the condition is examined when the supply flow is equal to 1.5 times the flow of one pump. The other operational possibilities — when the supply flow is less than the flow of one pump, and when the supply flow is greater than the flow of two pumps, must also be considered for a complete control strategy. In these examples, on and off can be synonymous with load and unload.

In **Operational Sequence A**, P1 operates continuously and P2 operates on and off in response to a level switch signal. The cycle time is obtained when sizing P2's storage or partial sump volume. In this operational sequence there are moderate flow variations.

In **Operational Sequence B**, P1 starts first, and then, at a higher level, P2 starts. Stop occurs simultaneously for both pumps at sump minimum level. The flow variation is very wide, but the storage or partial sump volume for P2 is reduced to almost half of that for Sequence A. Because of the common stop,

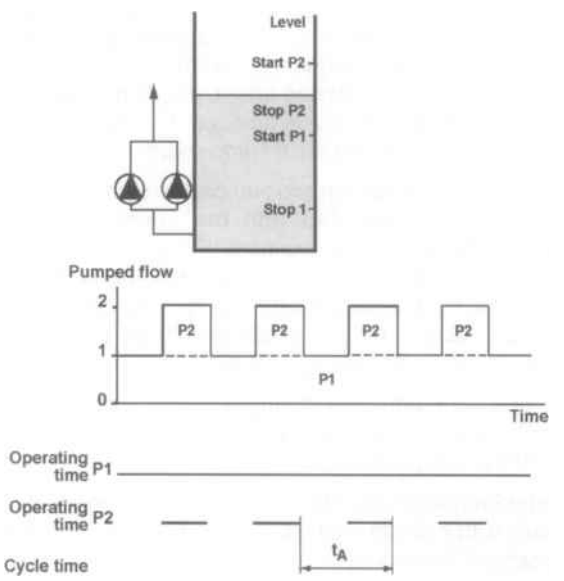

Figure 6.7 Flow variation and operating times for operational sequence A

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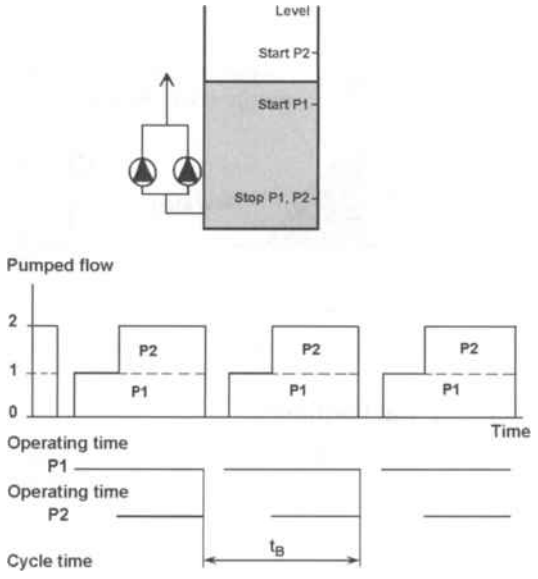

Figure 6.8 Flow variations and operating times for operational sequence B

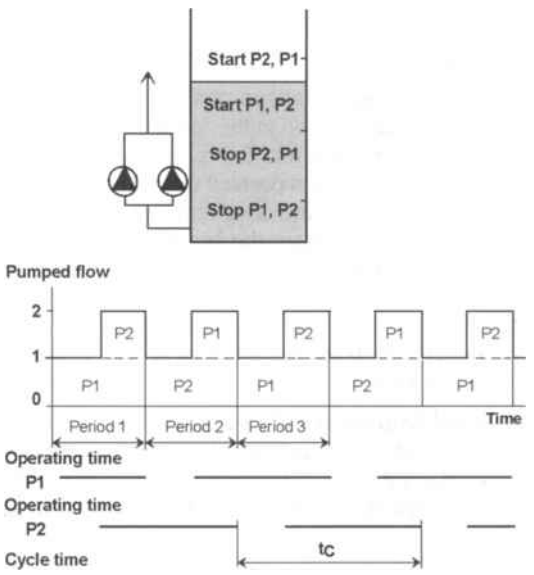

Figure 6.9 Flow variation and operating times for operating sequence C

one sensor signal is saved. Operational Sequence B has been applied to about 90% of all sewage pump installations up to the present time.

In **Operational Sequence C**, there is the same flow variation as in Operational Sequence A, except that the stop sequence is the same as the start sequence, i.e. the stop signal always stops the pump which started first. Operational Sequence C means a doubling of the cycle time, i.e. $t_C = 2_{tA}$, meaning the halving of the storage or partial sump volume for P2.

Where there are more than two pumps, the same principles as above are used, as also with the speed changing of pole-changing motors, for example. For a number of pumps working in parallel, the partial storage volume is then determined for each pump depending upon its respective incremental flow Q1, Q2, Q3, etc., as obtained from the pump and system curves, according to Figure 6.10.

In practice, to determine switching times and storage volumes, supply flows very close to pump capacities would be used, e.g. 99%, 101%, 199%, 201%, etc.

Powerful Programmable Logic Controllers, PLCs are currently available with multiple digital inputs and outputs and cost approximately $260; about the same as a good pressure switch. These devices allow on-off control to be completely optimised. For example, the PLC could keep a running total of the "hours

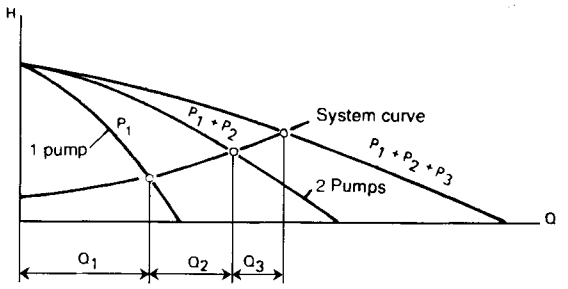

Figure 6.10 Flow increments Q1, Q2, and Q3 obtained by switching in various numbers of parallel operating pumps

run" for each pump and start the pump with the least hours first and switch it off last. A simple programme would try to balance the operating time across all the pumps. Operating times could also be biased so that one pump had few hours and could be considered as a stand-by.

When considering how many pumps to install, the possibility of failure and allowances for unavailability due to routine maintenance must be considered. The need for stand-by capacity must be weighed against the consequences of unavoidable reduced capacity.

6.4.6 Storage volumes

Storage and pump tank/sump volumes, see Figure 6.11, are most easily calculated by the use of a filling time, time constant, for the various partial volumes V, and V_2, and so on, defined as:

$$t_2 = \frac{V_2}{Q_2} \text{ etc.} \qquad \text{Equ. 6.6}$$

Time t_1 thus corresponds to the time taken to fill volume V_1 with flow Q_1.

In the case of an accumulator, Q varies because of counter-pressure from the pre-compressed volume of air or nitrogen. Boyle's Law at constant temperature: $p_1V_1 = p_2V_2$, should be used to calculate the change in gas volume with pressure. Time t_1 for the pump switched on first depends solely on the number of permitted starts per unit time, depending on whether starting is applied to several pumps alternately.

Time t_1 is determined by

$$t_1 = \frac{3600}{4 \cdot m} \text{ (s)} \qquad \text{Equ. 6.7}$$

where:

m = number of starts per hour per pump if the pumps are not alternated

or

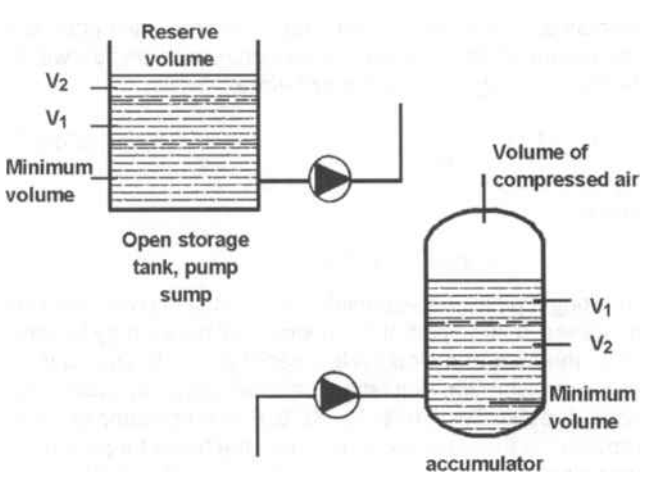

Figure 6.11 Storage with associated minimum volumes, partial volumes V_1 and V_2 and the volume of air for various arrangements

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m = number of starts per hour for the whole installation, if the pumps are alternated

Here the number of starts per hour is a maximum when the mean flow per hour is equal to half the maximum flow. The times t_2, t_3, t_4, etc., are heavily dependent upon the operational Sequences A, B and C.

Table 6.3 shows various values for up to four pumps operating in parallel. In the table it is assumed that no more pumps are included in each operational sequence range than are needed to satisfy the flow requirements. An example also follows.

Number of starts/hr per pump	5			10			15		
Sequence	A	B	C	A	B	C	A	B	C
Filling times for:									
pump 1: t_1	180	180	180	90	90	90	60	60	60
pump 2: t_2	180	72	90	90	36	45	60	24	30
pump 2: t_3	180	48	60	90	24	30	60	16	20
pump 2: t_4	180	40	45	90	20	22	60	13	15

Table 6.3 Table for determination of filling time for partial volumes in pump tanks/sumps

The air volume in an accumulator is calculated using Boyle's Law for isothermal compression, Figure 6.12.

At the absolute pressures p_{off} and p_{on}, there are respectively the volumes of air V_{Aoff} and V_{Aon}:

$$p_{off} \cdot V_{Aoff} = P_{on} \cdot V_{Aon} = P_i \cdot V_{Ai} \qquad \text{Equ 6.8}$$

where:

p_i = initial absolute pressure (kPa)

V_{Ai} = initial volume of air (m^3)

Using the effective volume of water $V_1 = V_{Aon} - V_{Aoff}$:

$$V_{Ai} = \frac{P_{on}}{P_i} \cdot \frac{P_{off}}{P_{off} - P_{on}} \cdot V_1 \qquad \text{Equ 6.9}$$

Combining equations 6.7 and 6.9, if only one pump of flow Q, delivers to the accumulator:

$$V_{Ai} = \frac{P_{on}}{P_i} \cdot \frac{P_{off}}{P_{off} - P_{on}} \cdot Q_1 \cdot \frac{3600}{4 \cdot m} \qquad \text{Equ 6.10}$$

where:

V_{Ai} = total volume of accumulator if not pre-charged (m^3)

V_{Ai} = volume of air in the accumulator when pre-charged (m^3)

p_{on} = absolute switch-on pressure (kPa)

p_{off} = absolute switch-off pressure (kPa)

p_i = absolute initial pressure (kPa)

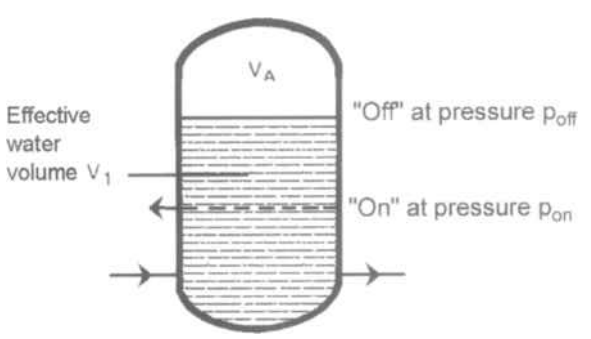

Figure 6.12 Accumulator with designations for working volumes

Q_1 = flow from pump, mean value of flow at p_{on} and p_{off} (m^3/s)

m = maximum number of starts per hour per pump (n/hr)

If several pumps are working on the same accumulator, the effective water volumes V_1, V_2, V_3, etc. are determined for each one as above. The various on and off switching pressures are then bound entirely by Boyle's Law, equation 6.8, which necessitates careful evaluation of the various quantities.

When liquid is supplied to processes which themselves are controlled by on-off valves the process control valves can create surges and water hammer effects. Large on-off process demands can have local accumulators to reduce peak loads on the supply system. The cost of local accumulators is offset by the reduction in supply pipe size necessary to feed the demand.

6.4.7 Power consumption

An on-off controlled pump operating at constant differential head has constant running power consumption, whilst the operational time varies with volume flow. The relative operational times, or factor of intermittency i_Q when transporting flow Q with a pump of flow Q_p will be:

$$i_Q = \frac{Q}{Q_p} \qquad \text{Equ 6.11}$$

The energy consumption E, i.e. the power requirement multiplied by time will be, for time t:

$$E = \frac{Q}{Q_p} \cdot P_{Qp} \cdot t \qquad \text{Equ 6.12}$$

In the case of varying flow the power consumption is summated for various intervals of flow. Simpson's Rule integration formula provides the total power consumption in simple terms thus:

$$E = P_{mean} \cdot t_o \qquad \text{Equ 6.13}$$

$$P_{mean} = \left(\frac{1}{6} \cdot \frac{Q_{100}}{Q_p} + \frac{2}{3} \cdot \frac{Q_{50}}{Q_p} + \frac{1}{6} \cdot \frac{Q_0}{Q_p}\right) \cdot P_{Qp} \qquad \text{Equ 6.14}$$

or at 5 points taken from the flow constancy curve:

$$P_{mean} = \left(\frac{1}{12} \cdot \frac{Q_{100}}{Q_p} + \frac{1}{3} \cdot \frac{Q_{75}}{Q_p} + \frac{1}{6} \cdot \frac{Q_{50}}{Q_p} + \frac{1}{3} \cdot \frac{Q_{25}}{Q_p} + \frac{1}{12} \cdot \frac{Q_0}{Q_p}\right) \cdot P_{Qp} \qquad \text{Equ 6.15}$$

where:

E = energy consumption (kWh/yr)

P_{mean} = mean power throughout the year (kW)

t_0 = operational time (hr/yr)

Q_0,Q_{50} = flow at 0%, 50% etc. constancy (m^3/s)

Q_p = flow at pump operating point (m^3/s)

P_{Qp} = pump power @ flow Q_P (kW)

For pumps operating at variable discharge pressure, with accumulators for example, consider the power consumption to be the mean of the two values at low and high pressure. Power can also be "wasted" due to throttling or modulated by-pass during starting and stopping. Pumps which run continuously but are loaded and unloaded will consume some power when unloaded. This energy consumption should be added to the power calculated from equation 6.15.

On-off control has good power economy when there is a high proportion of static differential head. The power economy may be expressed in terms of regulation efficiency η_r comprising the ratio between head requirement at each part of the system

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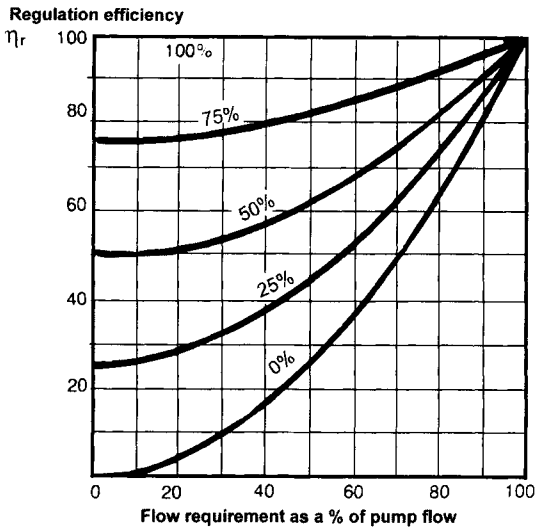

Figure 6.13 Regulation efficiency

curve and the pump head at flow Q_P. The regulation efficiency is shown in Figure 6.13 for on-off control of a single centrifugal pump working with water or liquid with characteristics like water for various ratios of H_{stat}/H_o. Ratio of the static delivery head as a percentage of total differential head H_o at the operational point of the pump. The total efficiency factor for liquid transportation is obtained by multiplying η_r by pump and motor efficiencies. See also Chapter 14, Section 14.3.1.

6.4.8 Examples

On-off control is used extensively in pumps for water, sewage and domestic drinking water installations. It is also commonplace in large hydraulic power packs used in steelworks and chemical processing. Some installations use 10 or 12 pumps to reduce the supply step size and also to limit the size of the liquid store. On-off control has many practical applications.

Certain special cases omit the liquid store, for example booster pumps, see Chapter 5, Section 5.3.3, working in "dead" pipeline networks. In order to avoid too frequent starting, an electrical interlocking device with, say, a time relay must be fitted. In certain flow ranges, pressure variations due to starting and stopping of pumps is inevitable.

In the case of circulatory pumping in totally enclosed systems, there is no requirement for a flow storage reservoir, but there is often a requirement for storage of some other parameter, e.g. heat in a thermal store.

Calculating storage volumes

The pump flows in Table 6.4 and storage volumes in Table 6.5, have been determined from the pump and system curves for a pump installation using three parallel connected pumps, see also Figure 6.10.

Number of pumps	Total Flow	Flow increment
1	100 L/s	100 L/s
2	180 L/s	80 L/s
3	240 L/s	60 L/s

Table 6.4 Pump flows

Number of pumps	Filling time	Partial storage volume
1	t_1 = 90 s	V_1 = 100 · 90 = 9000 L
2	t_2 = 45 s	V_2 = 80 · 45 = 3600 L
3	t_3 = 30 s	V_3 = 60 · 30 = 1800 L

Table 6.5 Storage volumes

From Table 6.3, there are 10 starts per hour using operational Sequence C;

The total storage volume will be
9000 + 3600 + 1800 = 14,400 litres ≡ 14.4 m³.

Added to this is a minimum volume and, for closed storage, also a volume of air, see Figure 6.11, as well as a storage volume dependent upon pressure surge, see Sections 6.4.3 and 6.4.4. The sensor signals for start-stop are arranged at levels or pressures corresponding to the various partial volumes. If operational Sequence A had been selected the storage volume would have increased to (100 + 80 + 60) - 90 = 21,600 litres ≡ 21.6 m³.

A storage volume for operational Sequence B = 100 - 90 + 80 - 36 + 60 - 24 = 13,320 litres ≡ 13.3 m³, is obtained in the same way.

If only the number of starts for each pump and not the total number of starts for the installation has been considered, there can be reductions in storage volume. With complete rotation using 1, 2 and 3 pumps respectively in operation and applying the principle of operational Sequence C, $t_1 = t_2 = t_3$ can be reduced to 30 seconds, giving a total storage volume of 7.2m³.

Furthermore, in many cases the partial storage volumes can, to some extent, be absorbed within each other, in which case, the minimum total storage can be estimated to be approximately 4m³.

It should be noted that a part of the cost saving for storage will then contribute towards more complex automation, although these costs are dropping dramatically with modern developments.

6.5 Pole-changing induction motors

6.5.1 Principle and application

Special versions of induction motors with arrangements for changing the number of poles can be used for pump control. Pump speed is inversely proportional to the number of poles, see Chapter 10.

Two distinct versions of motor are available with similar, but not identical, properties. Pole-changing motors achieve two or more speeds with one set of windings connected in different configurations. A dual wound motor has more than one set of windings; one for each speed. Small motors can have a switch built into the motor for local manual speed control. Usually all the winding terminations are brought out through the terminal box to the starter enabling remote automatic speed changes. Speed changing under load is normal. Star-delta (S-D) starting on all windings of dual wound motors is possible. With pole-changing motors, S-D starting is only possible on the lowest speed. Soft starts can be incorporated but the wiring becomes complicated. Pole-changing motors, and dual wound, can be used in those situations when a limited range of speeds; 2:1 or 4:3, 4:2; can be useful.

In some cases motor power is not always proportional to speed. Rotodynamic pump performance changes in accordance with the Affinity Laws, see Section 4.2.1.6, equation 4.16, in Chapter 4. Positive displacement pumps react differently to speed changes. The pump flow is approximately proportional to speed, depending upon the pump type and the viscosity of the liquid; the ability to generate pressure is unaffected.

Pole-changing regulation with rotodynamic pumps is particularly favourable when the system curve has only pipe friction losses as in Figure 6.14, in circulation installations for example. In this case the flow is approximately halved by switching to 750 r/min. Systems with H_{stat} will limit the value of the lowest useful speed.

Positive displacement pumps can work with systems with low and high H_{stat}. This is why these pumps are favoured in many chemical applications using constant pressure vessels and reactors. Gear, lobe, screw and progressing cavity pumps are

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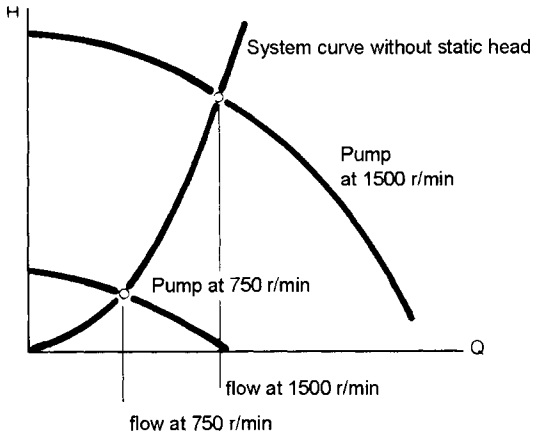

Figure 6.14 Example of change in duty point due to pole-changing 750/1500 r/min

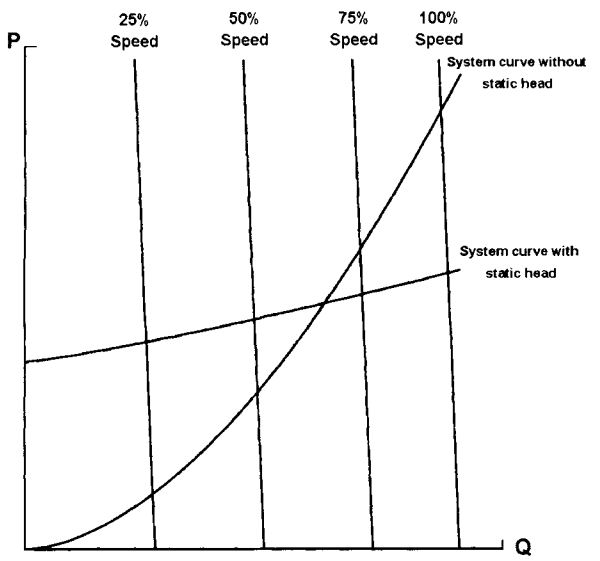

Figure 6.15 Positive displacement pumps at various speeds

used when viscosity is suitable; reciprocating pumps can be used for any viscosity. Figure 6.15 indicates the effect on the duty point of varying the speed of a PD pump when working with friction only systems and systems having a high H_{stat} value.

The power required by the pump is dependent upon the flow, the pressure and the corresponding pump and motor efficiencies at the duty point. Irrespective of the pressure, the flow varies very much in proportion to the speed change. The change in pressure is controlled by the system characteristic.

6.5.2 Costs

From the cost point of view, a variation factor of 2 is the most favourable since this gives pump speeds of, say, 750/1500 or 1500/3000 r/min, although other speed ranges are also possible. Some motors with four speeds are available, 1500/1000/750/500 r/min at 50 Hz. Pole-changing motors are cheaper than a motor plus a frequency inverter. If continuously variable speed is not necessary, and all-electric controls is an asset, the pole-changing motor should be considered. At 20 kW a two speed motor is 25% more costly than a standard motor, at 37 kW it is 50% more. At 135 kW a three or four speed motor is double the cost of a standard motor.

6.5.3 Starting problems

Starting problems are not as bad as those involved with fixed speed pumps with on-off or load-unload control because the pump can be started at low speed. Once the system has stabilised at low speed the motor can be switched to the next speed. The period of time required for stabilisation will depend upon the pump size and the system size.

6.5.4 Stopping problems

As with starting, stopping problems can be reduced by changing speed down in steps, with stabilisation periods, before switching off at the lowest speed. Power failure, when operating at the fastest speed, will cause the same problems as other systems. The reliability of the power supply, together with the likely problems, must be assessed when deciding if precautions are warranted, see Section 6.4.4.

6.5.5 Storage volumes

Since pole-changing regulation is a discrete method of regulation, liquid storage is necessary. Sizing is carried out in the same way as for on-off control, Section 6.4.6, with extra steps to allow for the speed increments. The permitted number of pole changes is usually between 1 and 15 times per hour but should be confirmed with the motor manufacturer once specific running conditions are known.

6.5.6 Power consumption

Rotodynamic pump and motor efficiencies are for practical purposes the same at the various speeds, which leads to good operational economy, see Figure 6.16. The efficiency of positive displacement pumps is dependent upon the discharge pressure, which is in turn, dependent upon the system characteristic. The cost of pole-changing motors is of the order of 20% to 50% higher than for normal, squirrel-cage induction motors. A limiting factor, to some degree, is the reduced availability of this type of motor from manufacturers.

6.5.7 Examples

Consider the pumps and system curves shown in Figure 6.10. It was shown in Section 6.4.8 that the nominal storage volume was 14.4 m^3. If the pump curves shown referred to a 4-pole pump, 1480 r/min, then it may be possible to consider pole-changing. Using the Affinity Laws, the pump curves are replotted at 6 pole speeds, 986 r/min. The closed valve head is greater than H_{stat} so the pumps will operate with the system curve successfully. The new values of Q_1, Q_2 and Q_3 become 60, 45 and 35 l/s.

Using the same switching per hour and operational Sequence C, the volumes V_1, V_2 and V_3 become 5400, 2025 and 1050 litres giving a total volume of 8475 litres. Pole switching produces a reduction in basic volume of 41%. Further reductions in volume, considering combining partial volumes as in Section 6.4.8, will lead to further reductions.

The flow range of the pump package has been increased from 240/100, 2.4:1, to 240/60, 4:1. Without considering mixed

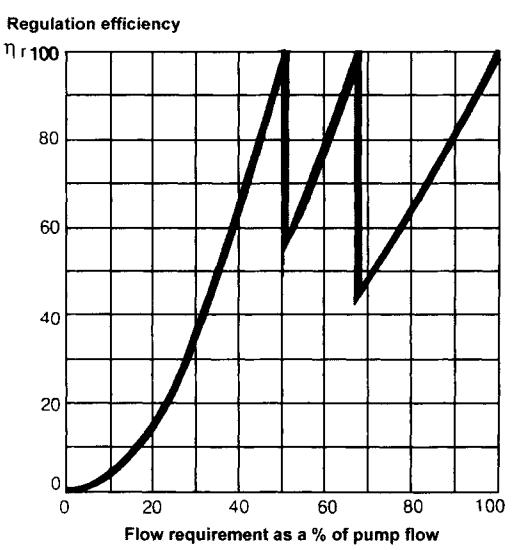

Figure 6.16 Regulation efficiency of a centrifugal pump using a pole-changing motor with 4, 6 and 8 poles. There is no static delivery head.

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speed operation, the operating capacities available are 25%, 42%, 44%, 58%, 75% and 100%.

The example from Figure 6.10 could not use pole-changing if the pumps had been running a 2-pole speeds. The next lower speed would be 4-pole and the pumps would not produce sufficient differential to overcome H_{stat}.

Pole-changing motors can offer increased flexibility and reduced storage volumes. It may be possible to run slow rotodynamic pumps in parallel with fast rotodynamic pumps; the relative shapes of the pump and system curves, together with the driver power installed, will determine whether this is feasible. Of course there are no problems running slow positive displacement pumps with fast positive displacement pumps.

6.6 Multi-speed gearboxes

6.6.1 Principles

The principle and application of multi-speed gearboxes is similar to that of pole-changing motors except there are no strict limitations on the speeds available. Manual change multi-speed gearboxes are nearly always built specially "to order" so the gear ratios can be optimised for the application. If a two pump system is considered, speeds of 100% and 50% could be chosen. This would provide pump package capacities of 25%, 50%, 75% and 100%. Again considering two pumps, gear ratios of 33%, 66% and 100% would produce package capacities of 17%, 33%, 50%, 67%, 83% and 100%. Two and three speed gearboxes are relatively common; more ratios are rare. Manual change gearboxes usually have internal dog clutches, the gears are always in mesh, for selecting the ratio. One or two levers are moved to change a ratio, while the drive is stationary. It may be necessary to jog or inch the motor to line up the dog clutches before engagement. On units up to about 250 kW, the high speed coupling can be turned by hand. On larger units facilities must be provided for barring, jogging or inching.

Semi-automatic gearboxes are frequently used on engine-driven portable pumps. Versions of these gearboxes are available for motor drives. Just like their automotive counterparts, gears can be changed onload without stopping. The gearbox is fitted with a torque converter or fluid flywheel to smooth the transition from one speed to the next. Remote and automatic control of gear changes is simple. The drive can be disengaged to give on-off control as well as variable speed. Larger units are fitted with a flywheel lock to save the 2 to 4% speed and energy loss due to flywheel slip. Gearboxes can be fitted with auxiliary power take-off to drive associated equipment. Standard gearboxes are available with up to ten gears. Choice of ratios is limited, but because of the multiple speed increments available this is not a problem.

6.6.2 Costs

For small manual change gearboxes, about 40 kW for a reciprocating pump, a two speed gearbox will cost approximately 4.5 times as much as a standard fixed ratio gearbox. A three speed box will cost 10 times as much. As the gearbox size increases the cost impact is reduced. For a 700 kW gearbox, two and three speed cost twice and 3.5 times as much.

Semi-automatic gearboxes start at approximately 28000 kW for an air operated four speed unit with fluid flywheel suitable for input speeds up to 3500 r/min.

The increased cost of the power transmission should be considered with the reduced costs of storage volumes. The overall scale of the pumping project will determine the relative cost proportions. The reduction in drive efficiency of semi-automatic gearboxes without fluid flywheel locks must be considered.

6.6.3 Starting problems

Manual change gearboxes are similar to pole-changing motors, see Section 6.5.3. Semi-automatic gearboxes, with more ratios, can have really low speed starts and smaller increments between the ratios effectively removing the problem.

6.6.4 Stopping problems

Again, manual change gearboxes are similar to pole-changing motors, see Sections 6.4.4 and 6.5.4. Semi-automatics do not experience problems with normal stops but power failures can be a problem, see Sections 6.4.4 and 6.5.4.

6.6.5 Storage volumes

Speed changing by fixed ratios is a discrete method of regulation and requires some form of liquid storage to compensate for the imbalance between supply and demand. Manual speed changing should only be considered when demand varies periodically rather than continuously. Manual speed changing can be viable if the demand only varies significantly say once or twice a day or less. At any given speed the pump should be considered as fixed speed and volumes calculated as shown in Section 6.4.6.

Semi-automatic gearboxes have no restrictions on gear changes and remote automatic control is relatively simple which includes disengaging the drive. The flows possible are in discrete steps, however small, and even with the additional on-off control some small storage capacity may be necessary. However, the storage capacity will be based on the smallest pump flow thereby reducing the storage required dramatically compared to manual speed change, on-off or load-unload options.

6.6.6 Power consumption

Power consumption will depend upon the curve shapes and the intersection points. Pump efficiency depends upon flow and head or pressure, driver efficiency depends upon the load. Manual change gearboxes are relatively simple so the mechanical efficiency should be about 95 to 97%. The basic efficiency of semi-automatic gearboxes will be slightly lower because of the greater complexity. Also the fluid coupling/fluid flywheel may lose up to 4% when not locked up.

Manual gear changing is very efficient at transmitting power but poor at changing gear. Semi-automatic gearboxes are good at changing gear at the expense of reduced transmission efficiency. The type and size of the pump package, limitations on storage volumes and cost, considered with the operational requirements will show which gearboxes are suitable.

6.6.7 Examples

Referring to the example shown in Figure 6.10, the minimum pump speed possible, allowing a 10% margin for wear and increased losses, is 60% of the speed shown. This would give a slowest running speed of 888 rpm based on an original 4 pole motor speed. Unusually, in this case $Q_1 = Q_2 = Q_3 = 40$ l/s. The basic storage volume would be 6.6 m^3. The flow capacity increments would be 17%, 33%, 42%, 50%, 75% and 100%. Using this pump on this system would produce a minimum safe capacity of 17%.

Positive displacement pumps do not have a closed valve head problem and any minimum speed could be selected to suit operations. Minimum speeds for reciprocating pumps can be as low as 5% of design.

6.7 Throttling by control valve

6.7.1 Principles

In the case of throttle regulation for rotodynamic pumps, Figure 6.17, the operating point of the pump changes because the system curve is modified. As the flow area of the control valve is re-

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duced the pressure drop increases, the flow through the pump reduces from Q_1 to Q_2.

Throttle regulation creates an extra flow loss $h_{f\,throttle}$ in the system. To overcome this, power must be delivered to the pump, thus:

$$P_{throttle} = \frac{\rho \cdot Q_2 \cdot h_{f\,throttle}}{1000 \cdot \eta_2} \quad (kW) \qquad \text{Equ 6.16}$$

In a completely zero loss regulation system, the power supplied would be less by $P_{throttle}$.

As can be seen from Figure 6.17, $h_{f\,throttle}$ and therefore $P_{throttle}$ are greater when the system curve is steeper for the same change of flow from Q_1 to Q_2.

Throttle regulation and by-pass regulation are the most common of the continuously variable methods of regulation, but throttle regulation cannot be used for positive displacement pumps. In view of rising energy costs, however, the method is becoming uneconomic despite the relatively low procurement cost of a control valve.

Throttle regulation is a continuously variable flow method. Operation is possible at any point on the system curve where the pump curve is higher. If the pump can be operated at closed valve without problems there is no requirement for storage volumes. The problem is efficiency; all the throttling energy is wasted. The choice of throttling is dependent upon the size of the pump, the cost of energy, how often throttling will be used and for how long. If the pump is small and/or energy is cheap throttling may be attractive. If throttling is only necessary for start-up and shut-down it may be attractive. Most pumps are installed with isolating valves. The throttle valve can be fitted as a by-pass around the isolating valve. After throttling for start-up the isolating valve can be opened fully for normal operation.

6.7.2 Costs

Installation costs are dependent upon the size of valve required and the mode of operation desired. Local manual control would be sufficient for start-up/shut-down that took one hour say every month. If continuous throttling was necessary a fully automatic control valve would be required. A 1 inch manual throttle valve, specifically designed for continuous operation would cost about $1500. A 2 inch low pressure constant flow control valve in cast iron suitable for water would be about $1000.

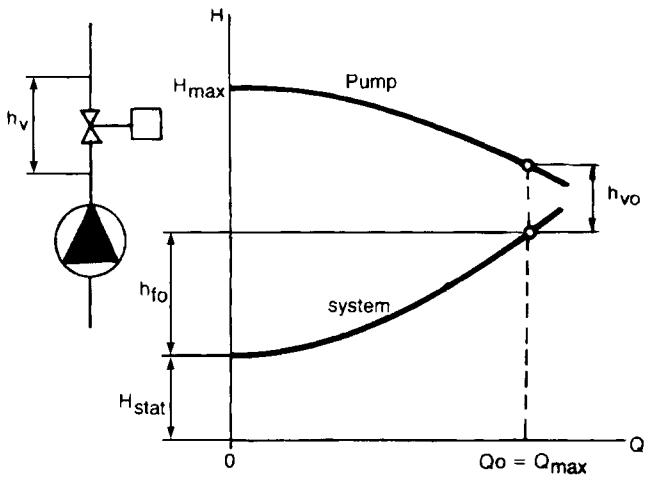

Figure 6.18 Relationship between pump and system characteristics for successful throttling

6.7.3 Sizing the control valve

The control valve is sized so that the control function itself has the best possible regulation characteristics. This means that the parameters to be controlled, i.e. the flow or the head, should be as linear as possible in relation to the movement of the valve. It is a requirement that a loss of head should always exist across the valve even at maximum flow Q_0, Figure 6.18.

When regulation is necessary in a system which has a simple smooth characteristic, as shown in Figure 6.18, the following approximation may be used for the minimum head loss at maximum flow:

$$h_{vo} = 0.1 \cdot h_{stat} + 0.3 \cdot h_{fo} \ (m) \qquad \text{Equ 6.17}$$

The sizing of the control valve and its linearity for this case has already been dealt with in Chapter 5, Section 5.2.4.

When the system curve is variable, throttling may be used to obtain a constant flow. The loss of head due to the valve must be selected at a considerably greater value than that given by the rule of thumb in equation 6.17. An analysis of regulation characteristics should also be carried out in the manner shown in the example in Section 5.2.4, of Chapter 5.

In throttle regulation, the power requirements of the pump are determined entirely by its power curve, regardless of what the

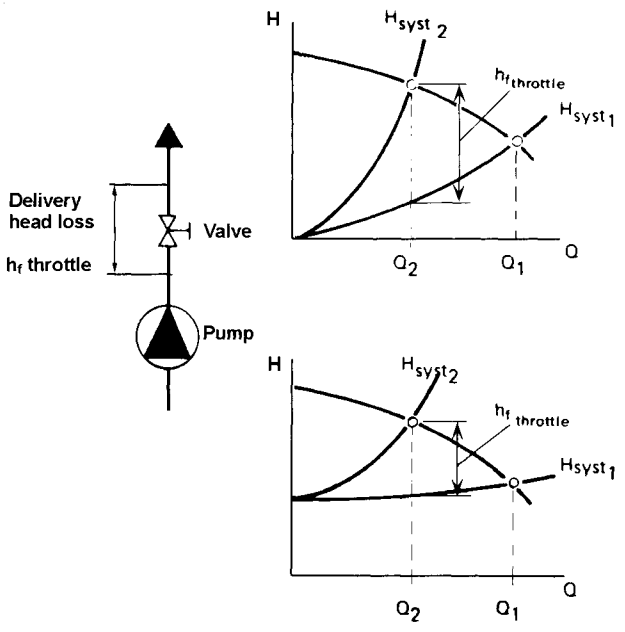

Figure 6.17 Throttling by control valve, schematic and curves

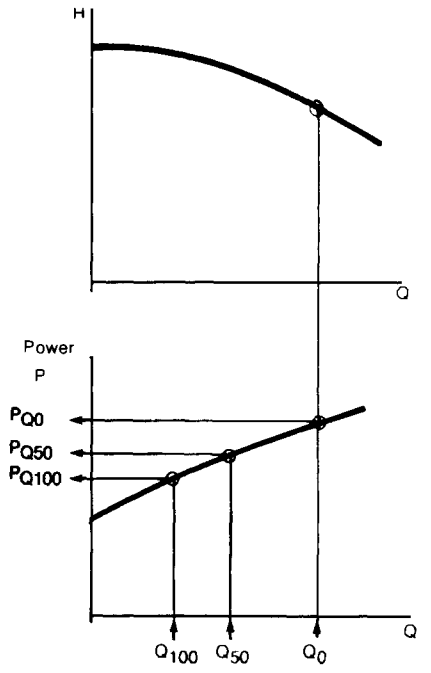

Figure 6.19 Pump power for throttle regulation

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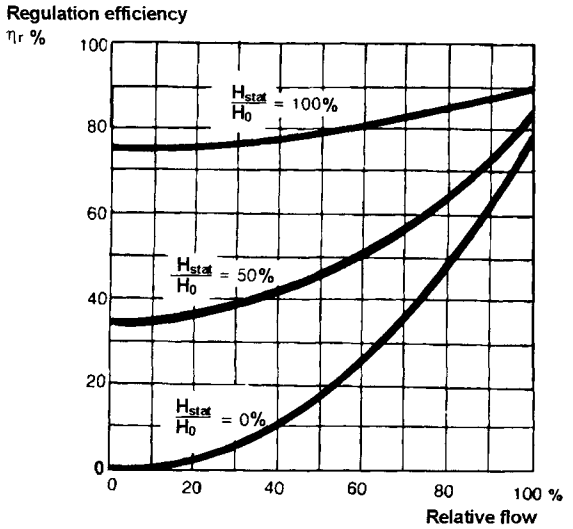

Figure 6.20 Regulation efficiency for throttling of a centrifugal pump with a shut-off head about 20% higher than the operating point at maximum flow

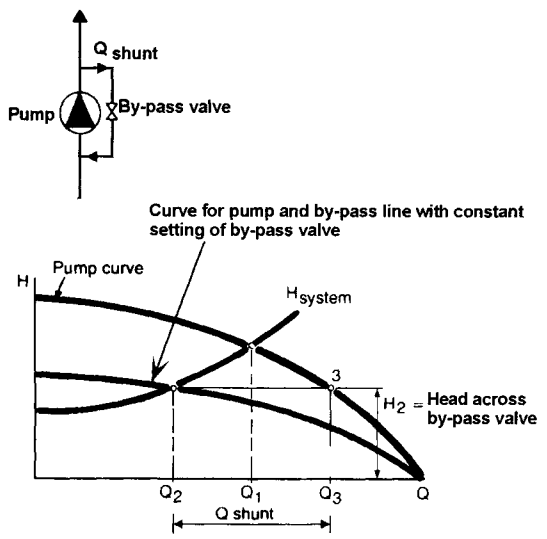

Figure 6.21 By-pass regulation, schematic arrangement and curves for centrifugal pump and by-pass line are reduced to a combined curve

system curve looks like, see Figure 6.19, i.e. the power requirement can be read directly from the pump power curve. In Figure 6.19 P_{Q0}, P_{Q50} and P_{Q100} are the power values corresponding to flow Q_0, Q_{50} and Q_{100} respectively, see Figure 6.22. These flows apply to mean flows at 100%, 50% and 10% constancy.

Energy consumption E is power multiplied by time, or

$$E = P_{mean} \cdot t_0 \qquad \text{Equ. 6.18}$$

where:

E = annual energy consumption (kWh/year)

t_0 = operational time per year (hr/year)

P_{mean} = mean power during operation (kW)

The mean power is most simply determined by the use of Simpson's Rule integration formula in the same manner as flow in Section 6.4, equations 6.1 and 6.2.

For throttling, the regulation efficiency is the ratio between the differential head of the system curve compared to the pump differential head. Figure 6.20 shows the regulation efficiency for the special case of a centrifugal pump working with water or a liquid with water-like qualities. The head H_0 applies to the head loss of the valve, i.e. $H_0 = H_{stat} + h_{fo}$, see also Figure 6.21. The loss of head due to the valve has been chosen in accordance with equation 6.17 and the pump curve is assumed to have its closed valve head some 20% higher than the operating point for maximum flow. In the case of throttle regulation, pump efficiency varies with flow and the motor efficiency varies with absorbed power.

6.7.4 Power consumption

Throttling increases the power consumption. The head loss through the control valve is completely destroyed but must be "paid for" with energy from the pump. If throttling must be used continuously it will be a continuous waste of energy. If throttling is only required for start-up or shut-down it can be accomplished in a by-pass around the isolating valve and once the isolating valve is opened the throttling loss will be zero.

6.7.5 Example

Consider the pumps and system shown in Figure 6.10. The process has been redesigned and only one pump is required. The storage volume has been moved to another part of the process. During start-up and shut-down, which last 2 hours each every month the flow through the system must be reduced to 50 l/s. The head drop through the valve must be 60 m. What size of valve is required ?

Using Equation 5.10 (in Chapter 5) with

$Q = 180\ m^3/h$ and $\Delta p = 60 \cdot 1000 \cdot 9.81/\ 100000 = 5.89$ bar

a minimum K_V of 74.2 is obtained. Reviewing manufacturers' data reveals the smallest valve suitable to be 80 mm, which would be at 80% open. This is considerably smaller than the pump discharge which is 150 mm. With the valve wide open K_V = 110, trying to pass 100 l/s would result in a pressure drop of 14.3 bar, 146 m. This pressure drop is higher than the closed valve of the pump so the pump would be unable to run at the normal duty point. This is an ideal case for putting the throttle valve in a by-pass around the discharge isolating valve.

6.8 By-pass return

6.8.1 Principle and application

By-pass regulation is sometimes called shunt regulation because of the similarity with shunts in electrotechnology. In practice, it is simply called by-pass. In by-pass regulation, as shown in Figure 6.21, liquid is fed back from the delivery side of the pump into the suction side of the pump. In practice, as with relief valves, the by-pass should be piped back to the suction source not to the suction pipe adjacent to the pump.

Operating points are most easily determined by combining the curves for a rotodynamic pump and the by-pass line into a single curve.

By-pass regulation is used for positive displacement pumps and only rarely for rotodynamic pumps. The energy consumption for these is generally greater with by-pass regulation than it is with throttle regulation. (See for example the pump curves in Figure 4.26, Chapter 4).

By-pass control is used with positive displacement pumps in three main modes of operation:

- Low pressure starting
- Load/unload control
- Small flow changes

Some positive displacement pumps operate in systems where the discharge pressure is almost constant, consequently starting takes place at high torque. This type of starting can generally only be accomplished with direct-on-line (DOL) starting. Not all electric supplies can cope with the current demands of DOL and star-delta is preferred, especially for motors over 15 kW. Star-delta starting does not provide 100% motor torque so that starting the pump on load is not possible. A by-pass, returning to suction source, provides a low pressure, low torque starting option.

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The pump starts with the by-pass wide open, the discharge pressure generated is only sufficient to overcome friction losses of the by-pass line and valve. Once the pump is running at full speed the by-pass valve is gradually closed, the discharge pressure increases until the non-return valve opens and the pump is online. The by-pass valve is completely closed and incurs no energy losses. The pump may be stopped by gradually opening the by-pass valve to reduce water hammer and pressure surges.

Step capacity control can be achieved on positive displacement pumps by loading and unloading the pump rather than starting and stopping by on-off. A by-pass valve is opened and closed and the pump flow varies from 100% to zero. The by-pass valve can be fitted with dampers to ensure long opening/closing times to eliminate water hammer and pressure surges.

When only small flow reductions are necessary a by-pass valve can be economically attractive compared to other methods. As only a small flow passes through the by-pass the amount of energy wasted compared to the pump power is small and only a small valve is needed.

Positive displacement pumps working with non-Newtonian liquids rarely have sufficient information to accurately predict system curves to allow the control characteristics to be analysed. Large safety margins are advisable. It should be noted that the control valve flow area may not be adequate for liquids containing larger particles. The power requirement is determined by the flow, $Q_{max} + Q_{shunt}$, and the discharge pressure according to the system curve at the actual system flow. Because of this, the regulation efficiency will be relatively high, Figure 6.22.

It is generally not necessary to have 20% extra capacity with positive displacement pumps. If the flow rate is critical then an allowance should be made for wear in the pump between overhauls. Depending upon the pump type, and how corrosive/abrasive the liquid is, this should be between 2% and 5%.

6.8.2 Costs

Costs for by-pass valves are similar to throttle valves. A 1 inch manual valve specifically designed to cope with the throttling and high speed liquid would cost about $1500. Valves are available up to 2.5 inches; a 1 inch valve has a C_v of 27, a 2.5 inch is 79.

6.8.3 Sizing the by-pass

The by-pass valve is sized in a similar manner to a throttle valve. Once the flow and pressure drop are known equations 5.10 or 5.11 (in Chapter 5) can be used to evaluate K_V or C_V. A slightly larger valve is not a problem; it just requires closing a little. A slightly smaller valve will stop the pump from operating down the system where it was intended.

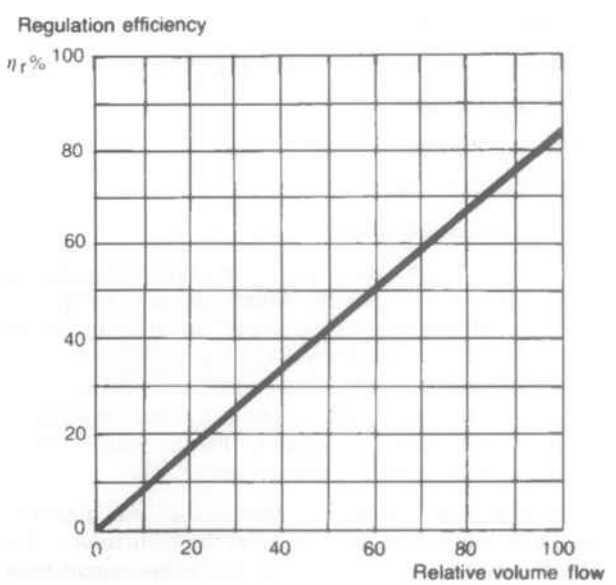

Figure 6.22 Regulation efficiency for by-pass regulated positive displacement pump with 20% by-pass at maximum required flow

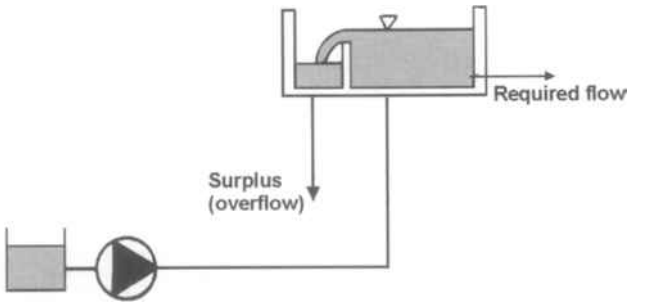

Figure 6.23 Overflow control

6.8.4 Overflow by-pass return

With overflow control the liquid is pumped into a reservoir at almost constant level. The required system flow is taken from the reservoir. The surplus liquid flows over a weir, which creates the constant head, and goes back to the suction side of the pump or escapes in to a drain pipe, Figure 6.23. The pump in such cases operates at constant flow and constant power requirement.

Overflow control presents a simple and reliable method but uses a relatively large amount of energy. The regulation efficiency depends upon the proportion of static delivery head, Figure 6.24. The figure makes the assumption that there are no unwanted level differences.

6.8.5 Power consumption

For rotodynamic pumps, energy consumption is determined as shown in Section 6.7.4, i.e. operating points on the H-Q curve are established, thereafter the power requirement of the pump is derived from its power curve. The regulation efficiency for rotodynamic pumps is generally lower with by-pass regulation than it is with throttle regulation, Figure 6.19. In spite of this, by-pass regulation of a rotodynamic pump may be justified because of the lower loss of head across the control valve with consequent lower noise levels and reduced risk of cavitation in the control valve. By-pass valves can also be closed completely so there is no energy wasted at all. The operating points of the rotodynamic pump should, however, always be checked for cavitation and abnormal vibration levels.

When a by-pass valve is closed there are no energy losses. When it is open the pump must provide the system flow and the by-pass flow, both at system pressure. The wasted power loss with by-pass regulation is:

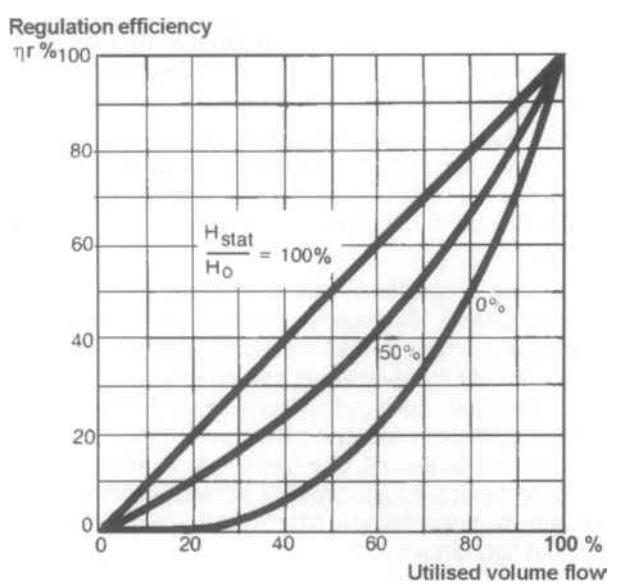

Figure 6.24 Regulation efficiency with overflow control, $H_0 = H_{stat} + h_{f0}$

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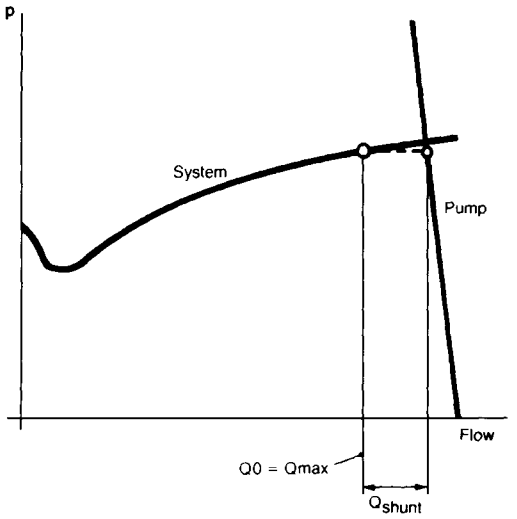

Figure 6.25 By-pass regulation of a positive displacement pump. The system curve shown applies to a non-Newtonian liquid

$$P_{shunt} = \frac{\rho \cdot g \cdot Q_{shunt} \cdot H_2}{100 \cdot \eta_3} \quad (kW) \qquad \text{Equ 6.19}$$

where η_3 is for flow Q_3 (in Figure 6.21).

For positive displacement pumps using by-pass regulation, the power consumption is determined by the back pressure, i.e. by the system curve. In the same way as for throttle regulation, the valve must be given priority in the system control loop. This is achieved by increasing the required flow, Figure 6.25.

6.8.6 Example

A fixed speed reciprocating pump operates in a system with a high static pressure. The nominal duty point is 190 m^3/h at 184 barg, the suction pressure is 1 barg. For starting purposes the system flow must be reduced to 100 m^3/h, which corresponds to a discharge pressure of 53 barg, what size of by-pass valve is required ?

Q_{normal} = 190 m^3/h, p_{normal} = 184 barg, Q_{start} = 100 m^3/h

The flow to be by-passed is 90 m^3/h and the differential pressure available is 131 barg. From equation 5.11 (in Chapter 5) the K_V of a suitable valve would be 7.86. Reviewing manufacturers' data, a 25 mm valve has the required K_V at approximately 85% opening. The extra capacity of the valve will allow for other losses in the by-pass system.

6.9 Infinitely variable speed

Infinitely variable speed provides infinitely variable flow. There are some fixed speed pumps which can provide infinitely variable flow. Reciprocating metering and dosing pumps are usually variable stroke and flow can be varied from zero to 100%.

For small flows metering pumps should be considered. Metering and dosing pumps can be surprisingly large. Plunger pumps can be built in variable stroke versions. These pumps were first developed as boiler feed pumps and pumps up to 75 kW were available.

For hydraulic applications, liquids with very good lubricating qualities, variable displacement vane pumps and variable stroke piston pumps are available. Some variable stroke piston pumps can actually reverse the flow direction, -100% to 0 to +100% flow, while running at constant speed. For hydraulic applications consider vane and piston pumps. Variable speed pumps will generally not offer zero flow. A minimum pump speed is required to assure proper lubrication of bearings. Variable speed pumps will tend to offer 5% to 100% or 10% to 100% maximum flow ranges.

The need to have infinitely variable speed to provide infinitely variable flow will ultimately depend upon:

- The liquid
- The size of the pump
- The type of pump
- The flow range required

6.9.1 Principles

Many pumping applications require the flow to be adjusted to several different values which sometimes cannot be accurately predicted. In these situations it is advisable to make the flow infinitely adjustable over a range. Changes in operating conditions due to fouling or corrosion in pipework, variations in process conditions or wear in the pump can be accommodated without compromising operational requirements.

With rotodynamic pumps, throttling and by-pass regulation could satisfy most of the needs. With positive displacement pumps, by-pass regulation could satisfy all of the needs. However, both throttling and by-passing are inefficient. Energy supplied by the pump to the liquid is destroyed in a valve, and may not be economically viable in larger installations or when used continuously.

Pump speed variation allows the pump capacity to be matched to the process demand. In Sections 6.5 and 6.6 pump speed was shown to be adjusted in discrete steps; pump capacity may or may not match the demand but it will be much closer than it would have been without speed adjustment. The ability to vary the pump speed infinitely allows pump capacity to match process demands exactly.

Pump speed can be varied in many different ways. Two main approaches are adopted; a fixed speed driver with a variable ratio transmission and a variable speed driver. The two approaches can also be combined.

6.9.1.1 Variable ratio transmissions

Some pumps, typically larger pumps, require some form of power transmission because the pump speed does not coincide with the driver speed. In these instances variable ratio can sometimes be incorporated in the speed reduction. The popular methods for varying speed ratios are:

- Mechanical variators
- Hydrostatic couplings
- Fluid couplings
- Torque converters
- Eddy current couplings

6.9.1.2 Mechanical variators

Mechanical variators are a sub-group comprising:

- Variable ratio belts
- Variable ratio chains
- Variable ratio friction drives
- Variable ratio gearboxes

Belt and chain drives usually have fixed diameter pulleys and sprockets. By adjusting the diameter of the pulleys/sprockets the speed ratio between the two shafts can be varied. Variable speed belt drives use special cone pulleys to adjust the diameters. Adjustment can be manual or automatic. Power transmitted is up to 25 kW and speed ranges of 9:1 are possible. Variable speed belt drives are mounted like normal belt drives, inside a guard provided by the purchaser.

Variable ratio chain drives are constructed like gearboxes, housed within a casing which acts as the oil reservoir. Fixed gear ratios can be incorporated to adjust the output speed. Some variable ratio chain drives are designed to be constant power transmissions rather than constant torque devices; this

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feature can be useful in some applications and must be considered in the cost evaluation. Small units are capable of 6:1 speed ranges, the larger units, 600 kW with a service factor of 1, only 3:1.

Variable ratio friction drives use discs, rollers or ball bearings clamped between two flat or specially shaped plates to transmit the power. One plate is mounted on the input shaft, the other on the output shaft. The variable ratio is created by adjusting the contact circles on the plates. Units up to 15 kW are popular; output speed ranges are about 6:1.

The variable ratio gearboxes described should not be confused with multi-speed gearboxes in Section 6.6. Variable ratio gearboxes are epicyclic gearboxes with small control drives attached to the outer gear annulus. The basic epicyclic gear is designed for the required full speed ratio. Then, by driving the outer gear annulus with a low power source, the gear ratio can be made infinitely variable. They are usually built specially "to order" so any configuration is possible.

6.9.1.3 Hydrostatic couplings

Hydrostatic couplings consist of a variable displacement pump and a fixed or variable displacement hydraulic motor mounted in one housing which acts as the oil reservoir. The fixed speed driver provides the power to the variable displacement pump; variable stroke piston pumps are most usual. The high pressure oil drives the output hydraulic motor. Hydrostatic drives are constant torque devices, full load torque is available at all speeds. Up to 200% instantaneous torque can be used at start-up. Speeds down to 0 r/min are not usual, most units have a minimum continuous speed limitation dependent upon the input speed to the unit. With 1450 r/min motors, the minimum speeds would be about 50 r/min. Hydrostatic couplings are very stable and have a linear characteristic at constant torque. Over-running protection is inherent in the design as is controlled deceleration. Output speeds up to the input speed are standard, 1900 r/min maximum, speed range 27:1 with powers up to 22 kW. Speed control can be adapted to suit the process requirements. Cooling is by natural convection around the housing, high ambient temperatures would restrict coupling capacity.

Hydrostatic couplings can be built as separate units. A power pack, with reservoir, motor, pump and cooler, can supply high pressure oil to a remote hydraulic motor. This format can be useful in hazardous areas. All the electrical equipment can be in a safe area with only the hydraulic motor in the hazardous area. Standard packages of this style are available for powers up to 150 kW. Specials can be built to order to cover virtually any speed and power requirements.

6.9.1.4 Fluid couplings

Fluid couplings are one of the two types of coupling which use oil to transfer power from the drive to the pump. Because these are "torque" transmitters they are always fitted in the drive train at the highest speed. Fluid couplings would be fitted in the input side of reduction drives and the output side of speed increasing drives.

The torque which is transmitted, and consequently the power, is dependent upon how much oil is circulating. A scoop in the oil reservoir adjusts the proportions of oil in the coupling to oil in the reservoir. When the coupling is full of oil 100% torque is transmitted. When used with centrifugal pumps on a pure friction system curve with no static head the fluid coupling is capable of speed ranges of about 8:1.

For constant torque loads, due to oil cooling limitations, the speed range is only about 4:1. At speeds less than 100%, the excess power is given up to the circulating oil and must be extracted by suitable cooling. The fluid coupling must always operate with some slip, consequently the maximum output speed is about 98.5% of the input speed. Typical input speeds are up to 1800 r/min. Couplings over 650 kW are available as standard. Larger units, up to 1.5 MW, use a pump rather than a scoop to control the oil flow. Some of the larger units have built-in gearboxes. Controls can be supplied to suit any system.

6.9.1.5 Torque convertors

Torque convertors are the rotodynamic equivalent of hydrostatic couplings. A centrifugal pump on the inlet shaft transfers energy to the oil. Inlet guide vanes direct the oil through an inward flow radial turbine on the outlet shaft. Torque convertors transmit torque and should be in the highest speed shaft. Torque convertors are almost constant power devices, rather than constant torque. At low output speeds output torque is increased. The output speed can also be increased above the input speed. One problem with torque convertors on manual control is that they can overspeed if the output load is suddenly reduced. Oil cooling is by external heat exchanger. Couplings have been supplied for over 10000 r/min and powers over 10MW are not uncommon. Some units have integral gearboxes. Controls can be fitted to suit any system.

A special hybrid consisting of a fluid coupling and a torque convertor is available in certain sizes. The internal controls select the appropriate drive mechanism to optimise efficiency.

Liquid couplings have an important attribute which is not related to their variable speed capabilities. All liquid couplings effectively isolate the two sections of the drive train. If a torsional analysis is necessary, the total drive train can be treated as two separate systems divided at the coupling. A liquid coupling will not transmit torsional vibrations and it will act as a dampener for cyclic torque and speed variations.

6.9.1.6 Eddy current couplings

Eddy current couplings are the electrical version of fluid couplings but can transmit more than 100% motor torque for starting. A small control current, typically less than 0.5 kW, adjusts the speed and torque relationship between the driven and the driving halves. Couplings are manufactured in standard sizes; the torque and speed range being a function of the driver/driven power to the coupling rating. Couplings are available up to 5MW.

6.9.2 Variable speed drivers

Variable speed drivers are a very diverse group with very different characteristics. All of the following are currently used:

- Petrol engines
- Diesel engines
- Gas engines
- Crude oil burning engines
- Air motors
- Steam turbines
- Gas turbines
- Turbo-expanders
- Hydraulic turbines
- AC squirrel cage induction motors
- AC slip ring motors
- Switched reluctance motors
- DC motors

The choice of driver is dependent partially on the size required, the speed range and the energy source available.

6.9.2.1 Petrol engines

Petrol engines are used mainly for portable applications, comprising both small light packages which can be man-handled and trailer mounted units. Speed adjustment is by throttle con-

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trol of the engine. The useful range of speed available is dependent upon the torque characteristic of the pump. Rotodynamic pump torque, for pumps operating on purely friction systems, varies approximately as the square of the pump speed, so allowance must be made for changing efficiencies. As pump speed is reduced from the rated condition, torque reduces quite rapidly. In these situations it may be possible to achieve a 4:1 speed range. For rotodynamic pumps operating with high static heads compared to friction heads the engine may struggle to cover a 2:1 speed range depending upon the pump characteristic.

Positive displacement pumps with frictional system losses may approach a 4:1 speed range; with constant discharge pressure systems the engine may have to be oversized to achieve any speed range if stalling is to be avoided. Small engines can operate up to 5000/6000 r/min, larger engines to 3000 r/min. A wide variety of sizes are available, up to 150 kW being popular.

6.9.2.2 Diesel engines

Diesel engines are used for fixed installations as well as portable machines. They are used on sites where electricity is not available or is not capable of supporting the pump load. All the comments on petrol engines regarding speed ranges apply to diesel engines. The speed available from diesel engines is lower; small engines about 3000 r/min, large engines down to 600 r/min. Small diesel engines of 5 kW are available, large fixed installations use engines up to 5 MW.

6.9.2.3 Gas engines

Gas engines are popular in fixed installations where natural gas is readily available. The natural gas can be used locally, usefully, rather than incurring additional cost by transportation or wasting by flaring-off. Full details of the gas specification must be available to the engine manufacturer to allow engine outputs to be predicted. Gas engines are more similar to diesel engines than petrol engines. All the comments on diesel engines apply to gas engines.

6.9.2.4 Crude oil burning engines

Like natural gas engines, crude oil burning engines are popular in fixed installations where there is a good source of supply and other energy supplies are limited or non-existent. Crude oil burning engines are similar to diesel engines. Some types of crude oil may require considerable pre-treatment before being suitable for use in the engine. Some engines may require starting and warming-up on diesel or gas before the crude oil can be used. Decisions and costing for crude oil engines can only be made when all the details of the actual crude oil are available. Extra costs incurred for pre-treatment or starting on other fuels may indicate other energy sources to be more cost effective. The comments for speed and speed ranges given for diesel engines apply to crude oil engines except that small engines are not available.

Dual fuel engines

As indicated in crude oil burning engines it is possible to run engines on more than one fuel, not generally at the same time. The most popular combinations are diesel and natural gas, or crude oil and natural gas. Full details of both fuels must be given to the engine manufacturer.

Turbo-charged engines

The details given in previous engine categories apply to naturally aspirated engines; that is engines which draw the air supply through an air filter from the atmosphere. Turbo-charged engines are becoming very popular. Turbo-charging significantly improves the engine efficiency and increases the power output without much change in the space required. In some cases turbo-charging seriously modifies the shape of the engine torque curve so that torque reduces rapidly either side of the optimum operating point. This change in engine characteristic can considerable reduce the operational speed range of any pump with any system. This feature must be considered, with all the other points, when deciding upon the type and size of an engine.

NOTE: The use of the waste heat, from the engine exhaust, can significantly improve the overall operating efficiency of the pump installation. The heat can be utilised for water or process heating or steam raising. In general, the extra cost of heat exchangers will be quickly recovered due to the large amount of heat available.

Engines for hazardous areas

Internal combustion engines can be modified to be suitable for operation in some hazardous areas. This requirement must be specified at the initial inquiry stage. Not all engines can be modified; not all manufacturers are interested in modifications. The surface temperature requirements, together with any relevant approvals, must be specified. Hot areas of the engine are cooled or insulated, electrical auxiliary equipment is uprated. Special attention is paid to air filters and silencers for arresting sparks.

Problems with engines in hazardous areas can be alleviated, in some cases, by installing the engine in a safe area separated from the hazardous area by a fire wall. Power transmission is via a spacer coupling which has a special mechanical seal, set in the fire wall, to prevent direct leakage of the hazardous substances.

NOTE: All engines can be used with multi-speed gearboxes to increase the overall useful speed range. The engine provides the infinitely variable speed capability and the gearbox can be used to provide several operational speed ranges. This approach should be considered when wide speed ranges are required.

6.9.2.5 Air motors

Pumps can be driven by compressed air. Air motors are available as standard assemblies. Torque characteristics should be almost constant so wide speed ranges are possible with all pumps. Sizes are limited to about 5 kW, speeds up to 9600 r/min are possible.

6.9.2.6 Steam turbines

Steam turbines are popular for driving larger pumps, but also small pumps in situations, such as refining and chemical processing, where steam is plentiful. Continuous processes are best so that time waiting for warm-up is periodic rather than regular. Steam turbines can be condensing or pass-out, that is to say the turbine expands the steam to a lower pressure which is then used in another part of the process. The torque characteristic is fairly constant, wide speed ranges are possible. Steam turbines are built from about 25 kW; there is no practical limitation on the largest size. Small turbines tend to run quite fast, 6000 r/min and faster, but there are no fixed speeds, the governor can be adjusted for a range of speeds.

6.9.2.7 Gas turbines

Gas turbines are useful for producing plenty of power from a small space; they also tend to produce much noise. The turbine itself can be small, but has a tendency to be overshadowed by the air filters and exhaust ducts. Gas turbines are ideal for driving fast pumps, speeds of 10000 to 15000 r/min are common. Slower pumps will require reduction gearboxes. Warming up is generally not a problem; full load can be available in 1 or 2 minutes. Gas turbines can utilise a variety of liquid and gaseous fuels. Crude oil burning is an option from two manufacturers.

NOTE: The usefulness as a variable speed driver is dependent upon the torque characteristic which in turn is dependent upon the gas turbine design. A single shaft machine, one in which the compressor and the turbine are mounted on one shaft, or two separate shafts but geared together at a fixed ratio, does not have an ideal

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torque curve. Below the design speed, torque reduces at a power slightly higher than a square law. The point of zero drive torque occurs at about 20% speed. This type of gas turbine will be suitable for some rotodynamic pumps whose system characteristics have no static head.

The other type of gas turbine is the free power turbine type; the compressor and its turbine are on one shaft, the turbine for external drives is mounted on a separate shaft and can rotate at virtually any speed required. The torque curve of this gas turbine is ideal. The compressor can be allowed to run up to full speed, if necessary, before the pump is started. Starting torque can be over double the normal running torque, reducing almost linearly to full load torque at full speed. Any type of pump or system can be operated with a free power turbine gas turbine. Speed ranges over 5:1 are possible. Turbine cooling may be a problem for low speed continuous operation with constant torque pumps.

Gas turbines start at about 25 kW. Overall efficiency can be improved by utilising the waste heat in the exhaust gases. It is more cost effective to design the pump/system around a standard gas turbine than to ask the turbine manufacturer to modify the turbine to suit the pump/system. The world supply of gas turbines, in some sizes, is extremely limited. When discussing operational requirements for gas turbine drivers also ensure that machines will be available for delivery to suit the project timescale. Some projects have been postponed for two years waiting for turbine availability. Several manufacturers operate allocation systems and it is essential to reserve a place in the queue.

6.9.2.8 Turbo-expanders

Turbo-expanders are effectively the power turbines of gas turbines without the gas generator. They can be used where a supply of gas is available which can be expanded. The gas utilised may be process gas which is expanded between process stages or waste gas which is expanded to atmospheric conditions. Speed and torque curves can be adjusted by control and by-pass valves; most pump requirements can be accommodated. They are built in standard sizes which can be adapted to suit individual conditions. Turbo-expanders are also built specially "to order" when a sufficient number of machines is required.

6.9.2.9 Hydraulic turbines

Pumps can be driven by hydraulic turbines when a source of high pressure liquid is accessible. The turbine can be a Pelton Wheel, a Francis turbine or a pump running in reverse. The pump can be axial flow or radial depending upon the head and flow available. All hydraulic turbines can be controlled to produce variable speed outputs. The choice of turbine type will depend upon the characteristics of the driven pump system. Pumps running in reverse have become popular over the last 10 to 15 years. Many parts can be made common to the pump and only one manufacturer is involved. Size and speed is virtually unlimited.

6.9.2.10 AC squirrel cage induction motors

A standard, off-the-shelf motor, driven by a variable frequency inverter can produce speed ranges of 10:1, and over, without problems. This method of speed control is the most popular of all methods for motors up 125 kW, perhaps even 250 kW.

Full load torque, plus a margin, is available for starting without transmitting large current demands into the supply system. Full load torque is available over the whole speed range. Some motors may need derating by up to 10% to work with some types of inverters. There is a need to match the frequency inverter to the motor in the larger sizes. Care should be taken with continuous operation at low speeds. Some motors may need a separate motor driven cooling fan.

Discussions should be held with the motor and inverter manufacturers regarding operating conditions. Special attention may be needed with starting torque or overload capabilities for operating at relief valve settings. If accurate speed control is required, for metering pumps for example, some feedback from the motor to the inverter may be necessary. Some small inverters can accept single phase power at 240 V and drive three phase motors at 380 V. Some inverters can produce output frequencies up to 120 Hz.

The motor manufacturer must be consulted if motors are to be overspeeded on a regular basis. Virtually any control system can be interfaced to an inverter. Large inverters can transmit significant harmonic signals into the mains supply. Generating authorities should be consulted when contemplating drives over 500 kW. All inverter drives must comply with the statutory requirements for electromagnetic pollution in all countries.

6.9.2.11 AC slip ring motors

Slip ring motors are also called AC wound rotor motors. Slip ring motors were the standard before squirrel cage motors were invented. The lack of slip rings, plus simpler starting, made squirrel cage motors much more attractive for most fixed speed applications. However, slip ring motors continued to be used for limited variable speed applications and those applications which involved large inertias requiring long run up times.

The external resistance, connected in series with the rotor windings, allows the speed/torque characteristic of the motor to be modified. At low resistance, when the slip rings are shorted, the torque characteristic is very steep giving a wide range of torque values for small variations in speed. As the rotor resistance is increased the speed/torque characteristic becomes shallower giving corresponding torque values at lower speeds. Full load torque can be made available at any speed. This method of speed variation can be applied to most practical pump installations. However, it is not efficient. At reduced speeds, energy is dissipated in the rotor resistances and external cooling, motor driven fans, may be necessary. The motor efficiency reduces approximately in proportion to the motor speed.

The sliprings and brushes of wound rotor motors require maintenance and this is a disadvantage compared to squirrel cage motors. Problems can arise with hazardous area requirements. AC slip ring motors can be useful for large installations, with heavy inertias creating starting problems, when speed variation is only required periodically. If a pump was also required to run as a turbine, then the slip ring motor could become an alternator and produce power.

6.9.2.12 Switched reluctance motors

The switched reluctance motor, srm, is not a new motor design but an old motor design concept which has become viable with the advances of modern electronic power switches. The srm is similar in some respects to the squirrel cage motor. The motor rotor does not require copper or aluminium conductors and can be made from a single steel forging. Very high motor speeds are possible. The power electronics are used to switch the supply current to different stator windings. A srm installation is similar to a variable frequency inverter supplying a squirrel cage motor; the srm "inverter"' is much simpler than the squirrel cage version.

The srm package is very useful because of the extra features possible compared to the inverter/squirrel cage package:

- 100% torque at zero speed
- 100% torque holding at zero speed
- No motor overheating at low speed
- No motor overheating due to starting inrush currents
- Precise speed control

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- Very high speeds possible

The srm does not have a synchronous speed and cannot be operated without the power electronic controls. The switched reluctance motor can be expected to produce significant changes in the ways variable speed rotodynamic and positive displacement pumps are applied. The very low speeds possible should result in very small accumulators being fitted rather than large storage tanks. Specialist motor manufacturers are concentrating on motors up to 300kW.

6.9.2.13 DC motors

DC motors were the principal electrical variable speed drivers before variable frequency inverters became economically viable. Most applications which used to have DC motors can now be driven by variable frequency AC motors. DC motors have two main advantages over AC motors; they can be designed to run at any particular speed and have a faster response to transient conditions.

DC motors do not have synchronous speeds like AC motors. A DC motor could be designed to run at 4500 r/min, for example, to eliminate the need for a gearbox. If rapid speed changes are required as part of the normal operating conditions then DC motors may be better than AC. However, consideration should be given to the flow variations and pressure pulsations caused by any rapid speed changes.

The speed of a DC motor, up to its base speed, is controlled by the armature voltage. When the armature current equals the rated current, full load torque is available. These motors can provide full load torque from zero speed up to base speed. Speeds above base speed, produced by field weakening, are at reduced torque. Above the base speed, DC motors are constant power, not constant torque. Motor efficiency is high over the whole speed range. Modern motors are powered via electronic controllers in a similar manner to variable frequency AC motors. DC motors have commutators and brushes which require maintenance attention similar to sliprings. Hazardous area requirements can cause problems. DC motors are not as popular as AC motors and are not mass produced in the same quantities.

6.9.3 Costs

Procurement costs for various speed regulation equipment depends upon the shaft power, maximum speed and the environment in which the drive is going to be located. To some degree, extra allowances must be made for ancillary cooling equipment and in the case of electrical methods, of the effects of feedback in the mains from thyristors, and protection against incoming mains transients. The method of speed regulation selected may affect the procurement costs by a factor of 5. Figure 6.26 indicates the cost of standard three phase squirrel cage AC motors. Figure 6.27 indicates values for additional cost for speed regulation in relation to the costs for standard constant speed squirrel cage induction motors. For choice of equipment see Section 6.10.

6.9.4 Conversion of pump curves to various speeds

For rotodynamic pumps the Affinity Laws govern performance at speeds n_1 and n_2. If equation 4.16, in Chapter 4, is rewritten for constant diameter impellers then:

$$\frac{Q_2}{Q_1} = \frac{n_2}{n_1}$$

$$\frac{H_2}{H_1} = \left(\frac{n_2}{n_1}\right)^2$$

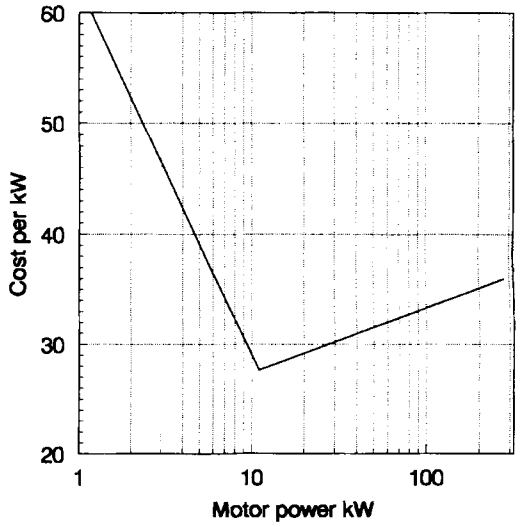

Figure 6.26 Costs of standard three phase squirrel cage AC motors

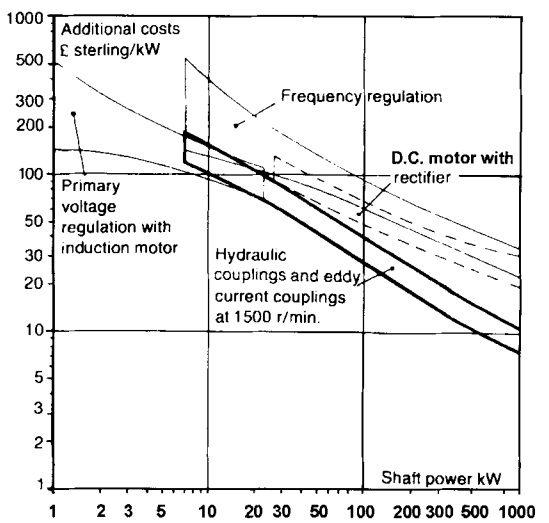

Figure 6.27 Additional costs for speed regulation for standard constant speed squirrel cage AC motors

$$\frac{P_{p2}}{P_{p1}} = \left(\frac{n_2}{n_1}\right)^3 \qquad \text{Equ 6.20}$$

The third part of the equation for power applies only to the absorbed power of the pump P_p.

NOTE: The Affinity Laws are used to modify the pump characteristics. The Affinity Laws do not predict how a pump in a system will operate. The equilibrium of the pump and system characteristics must be found in order to calculate the pump power and efficiency.

For any given pump in any given system, in accordance with Figure 6.28, the minimum required speed, n_{min}, can be calculated from the following equation:

$$n_{min} = n_{max} \cdot \sqrt{\frac{H_{stat}}{H_{max}}} \qquad \text{Equ 6.21}$$

The lowest required speed has to be known when selecting speed regulation equipment.

The performance of speed regulated pumps is usually described as a family of curves for H-Q and Q-P as shown in Chapter 5, Figure 5.23. These curves were determined by means of the Affinity Law, equation 6.20. The power requirement in accordance with equation 6.20 refers to the power requirement at the pump shaft. The power requirement for the electric motor is obtained by dividing by the efficiencies of both the motor and the transmission, see also Figure 6.29. Frequency inverters and thyristor controllers for DC motors must

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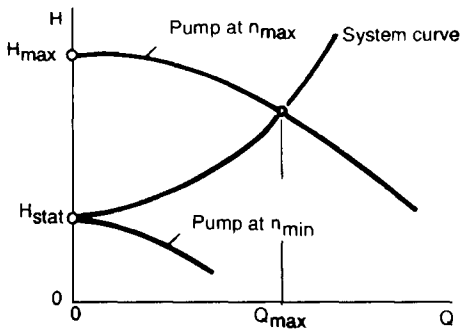

Figure 6.28 Relationship for lowest required speed, n_{min}

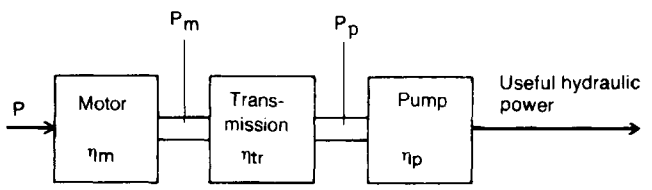

Figure 6.29 Pump unit with power transmission

be included. The overall efficiency is also called the "wire to water" efficiency.

For positive displacement pumps the flow is approximately proportional to speed, $Q \propto n$. The first part of equation 6.20 can be used but some allowances may be necessary for variations in slip. Slip will vary with speed and differential pressure. The differential pressure must be evaluated from the system characteristics, see Figure 6.15.

Pump power is the product of flow and differential pressure divided by the efficiency. Just as with rotodynamic pumps, pump efficiency varies with operating conditions.

Thus for rotodynamic pumps, drive power P:

$$P = \frac{P_m}{h_m} = \frac{p_p}{h_m \cdot h_{tr}} = \frac{r \cdot Q \cdot DH \cdot g}{h_m \cdot h_{tr} \cdot h_p \cdot 1000} (kW) \qquad \text{Equ 6.22}$$

For positive displacement pumps with flow in m^3/h and differential pressure in bar, drive power P:

$$P = \frac{P_m}{h_m} = \frac{p_p}{h_m \cdot h_{tr}} = \frac{Q \cdot DP}{h_m \cdot h_{tr} \cdot h_p \cdot 36} (kW) \qquad \text{Equ 6.23}$$

6.9.5 Efficiencies of various methods

If the Affinity Laws are applied to a centrifugal pump then, when the pump speed is regulated down to half speed, the volume flow also reduces by a half, the differential head to a quarter and the power required at the pump shaft to an eighth based on a system without static head. Similarly, when regulating down to 20% of maximum speed the pump shaft power goes down to 0.8% of the power at maximum speed based on zero static head. The variable speed pump characteristics must be plotted against the system curve to evaluate the speed, torque, efficiency and power for each duty point. At low speed, losses within the pump, such as bearing friction and seal friction, may distort the power curve so that the Affinity Laws underestimate the power required.

For positive displacement pumps the procedure is exactly the same. The pump characteristic shapes will be different as will the variation in pump efficiency. The pump torque and power for any duty point will be dependent upon the intersection with the system curve.

The characteristics of variable speed devices differ considerably. Not only is it essential to know the speed, torque and power of each duty point, but it is essential to know the timescale of operation at each duty point. It does not make sense to purchase a costly system which provides high efficiency at 10% flow if the only time low flow is necessary is during start-up and shut-down which occurs twice a year for a couple of hours.

Figures 6.30 to 6.34 indicate the general characteristics of the popular methods used for speed variation. The actual speed range possible is dependent upon the actual torque required at any speed. Efficiencies for drivers do not include the efficiency of any speed reduction necessary. Gearboxes should have transmission efficiencies of 95 to 98%. Efficiencies for variable ratio transmissions do not include the driver efficiency. Electric motors, the most popular driver, will have efficiencies between 80 and 96% depending upon the size.

The two curves in Figure 6.34 represent the variation in efficiency due to size. The dotted lines indicate a possible reduc-

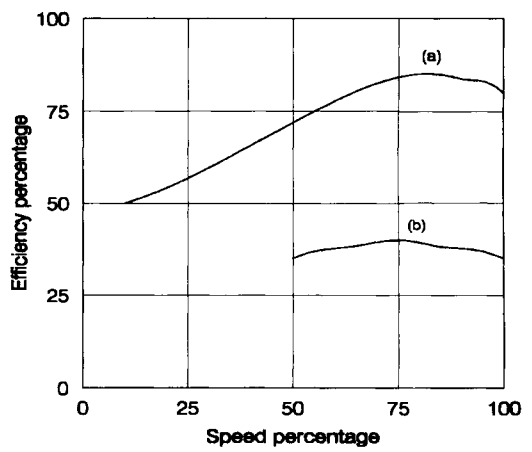

Figure 6.30 Speed range and efficiency — (a) torque convertor (b) engine

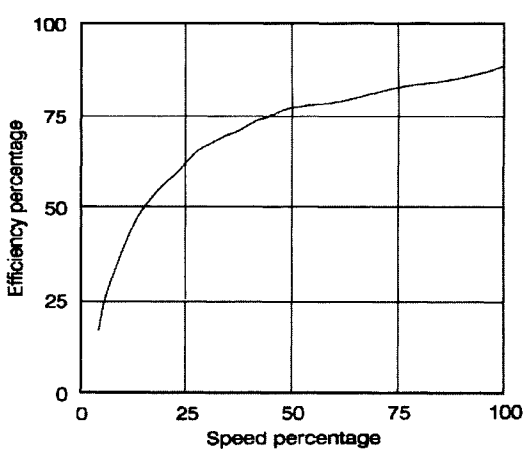

Figure 6.31 Speed range and efficiency of hydrostatic transmission

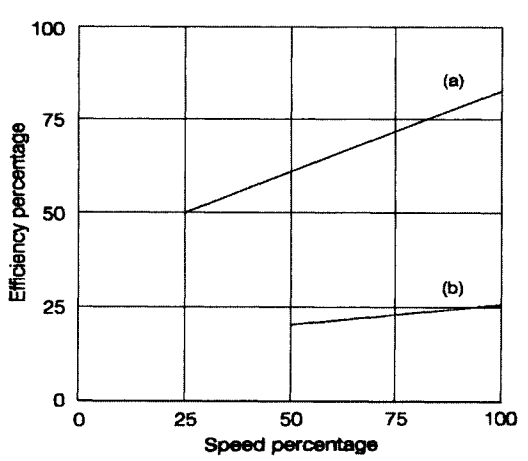

Figure 6.32 Speed range and efficiency — (a) steam turbine (b) gas turbine with free power turbine

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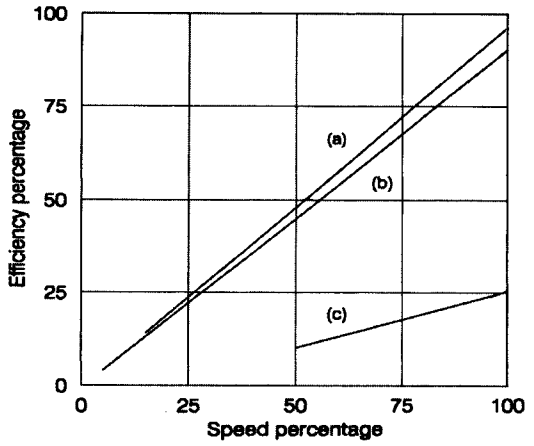

Figure 6.33 Speed range and efficiency —

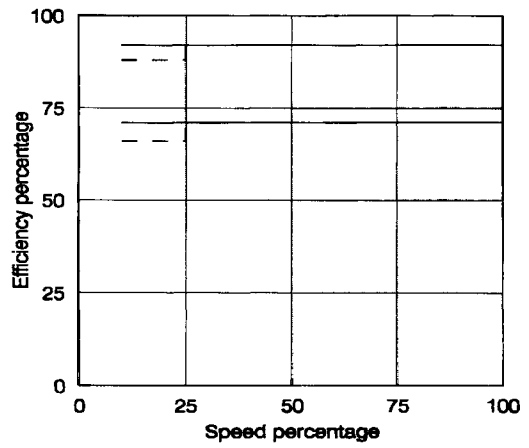

Figure 6.34 Speed range and efficiency of electric motors

tion in efficiency at low speed due to the requirement to start a separate motor driven fan to cool the main motor. Any power required for cooling fans fitted to electronic control equipment must be considered in the overall efficiency assessment.

6.9.6 Power consumption

In recent years, users whose operating costs are dominated by pump energy consumption have pioneered the concept of 'wire to water' efficiency. The pump and drive train efficiencies are not considered in isolation but are combined to produce an overall efficiency. This process includes variable frequency inverters for AC motors and thyristor controllers for DC motors. Some users have expressed concern over the accuracy of power measurements, when using the pump manufacturer's standard instrumentation, for electronic motor speed controllers. The controllers distort the power supply by injecting harmonics. Some users have insisted that the pump manufacturer needs modern, costly, analysis equipment to accurately measure the power consumption. This may in fact be strictly correct and theoretically possible. However, the pump user will probably be charged for electricity based on measurements taken by an old-fashioned kWh meter. This then, is the way to measure efficiency when the pump is being tested.

In operation, of course, the performance of a speed-regulated pump follows the requirement of the system curve in the H-Q diagram. By transferring the intersection points of the system curve with the various pump curves on the H-Q diagram to the power diagram the dotted line curve in Figure 6.35 is obtained. This curve indicates the power requirement when operating along the system curve. The power requirement is for a pump with fluid coupling. The speed is expressed relative to maximum speed for a direct drive pump no = 1. Because of slip in the fluid coupling a slightly lower maximum speed pump curve is produced compared to an equivalent direct drive pump.

The energy consumption E in kWh/year will be the power multiplied by time, or

$$E = P_{mean} \cdot t_0 \qquad \text{Equ 6.24}$$

where t_o = operational time per year (hrs) and where the mean power is determined by using Simpson's Rule:

$$P_{mean} = \frac{1}{6} \cdot P_{Q100} + \frac{2}{3} \cdot P_{Q50} + \frac{1}{6} \cdot P_{Q0} \quad (kW) \qquad \text{Equ 6.25}$$

The values P_{Q100}, P_{Q50} and P_{Q0} are the values for power requirement taken from the system curve at volume flow Q_{100}, Q_{50} and Q_0 respectively, i.e. the flow at 100%, 50% and 0% constancy. See also Figure 6.36. Taken at various values of Q provides the basis for determination of mean power in accordance with equation 6.25.

In speed regulation, the regulation efficiency is always equal to 1 because supply always equals demand. The losses in speed regulation are instead described by a transmission efficiency η_{tr}, see also Figures 6.29 to 6.34. The variations in pump speed

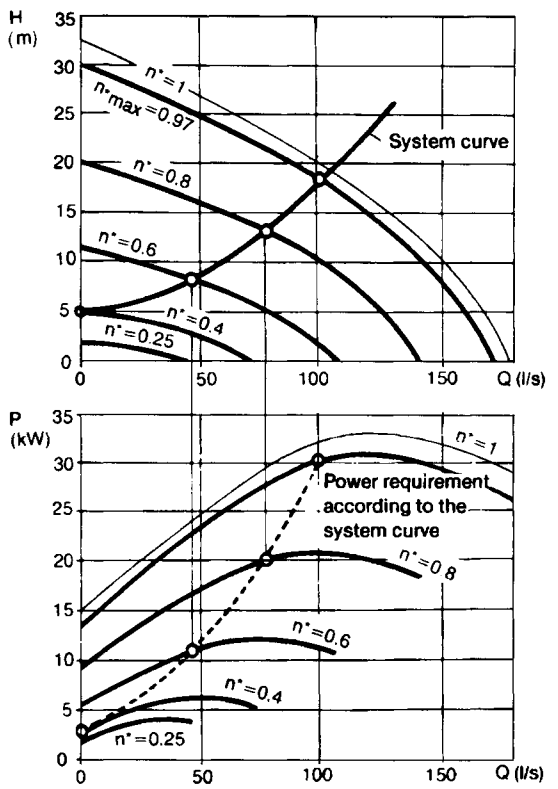

Figure 6.35 Example of performance curves for a speed regulated pump

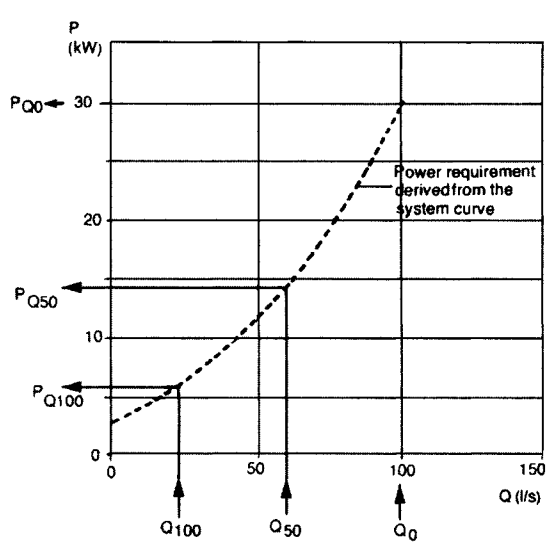

Figure 6.36 Power requirement for speed regulated centrifugal pump in accordance with Figure 6.35

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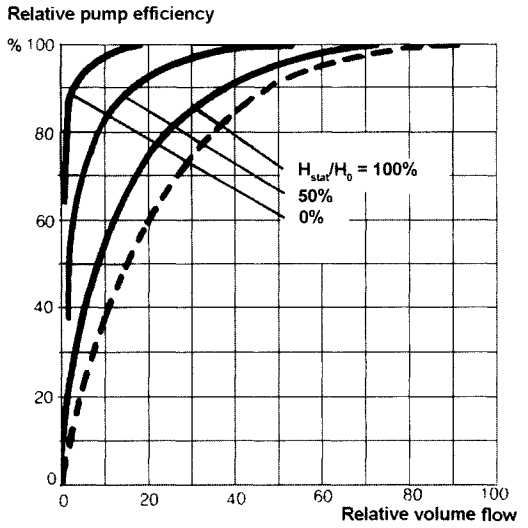

Figure 6.37 Examples of pump efficiency variation for a speed regulated centrifugal pump

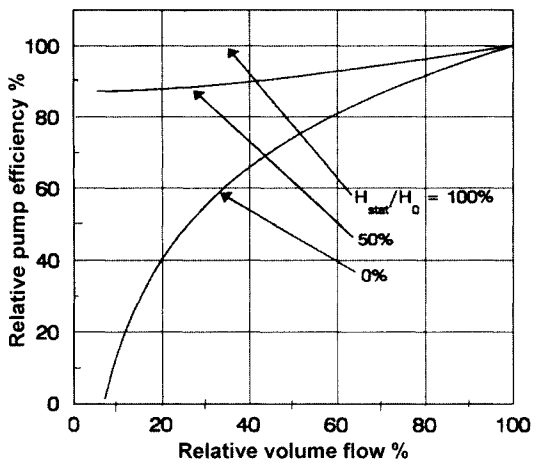

Figure 6.38 Examples of pump efficiency variation for a speed regulated reciprocating pump

and also transmission efficiency will depend upon the shape of the system curve. In some drive arrangements, AC motors with low speed pumps for example, transmission efficiency must include everything but the pump.

$$h_{tr} = h_i \cdot h_m \cdot h_g \qquad \text{Equ 6.26}$$

where:

η_i = inverter efficiency (decimal)

η_m = motor efficiency (decimal)

η_g = gearbox efficiency (decimal)

The transmission efficiency for a large drive train would be

$$\eta_{tr} = 0.97 \cdot 0.96 \cdot 0.97 = 0.903$$

Rotodynamic pump efficiency varies more favourably when speed regulated than with other methods of regulation, Figure 6.37. The dotted curve represents the normal efficiency change for a constant speed pump when 100% flow is at BEP. With frictional losses only, (H_{stat} = 0), pump efficiency remains constant over the larger part of the flow regulating range. It should be noted, that if H_{stat} is not equal to zero, then better part-load efficiencies are obtained if the operational point of the pump at full load lies to the right of best efficiency point. Some process users will not relish operating passed BEP and detailed calculations will be required to confirm the financial benefits.

Reciprocating pumps, not surprisingly, react differently to system variations. As static head increases in a friction system a reciprocating pump is able to operate closer to its optimum rod load even as flow is reduced. The best system is one where the discharge pressure is constant irrespective of flow rate, see Figure 6.38.

6.9.7 Regulation system schematics

A schematic showing the principle of operation for the complete regulation process of a speed regulated pump, by variable ratio transmission, is shown in Figure 6.39. The control system can be local or more usually a sub-system in the plant control system. Linearity between signals and required pump function is of the utmost importance for functional stable regulation. This condition is emphasised particularly in public water supplies where very great variations of flow are common. In this application, unfortunately, a number of installations using speed regulation are unusable for the simple reason that the importance of the regulation function was not fully appreciated. In many cases it is necessary to measure additional parameters, e.g. pump speed, n, or flow, Q, (dotted signals in Figure 6.39) and to use these as feedback into the regulation system, if stability is to be achieved.

Linearity is exploited through comparison between the required control parameter, generally flow, Q, or speed, n. If both are measured, the control system can compensate for wear and report on pump condition. When H_{stat} is not equal to zero, Q and n are no longer linear, see Figure 6.40. For rotodynamic pumps operating at high differential heads Q and n will never be linear. As can be seen from Table 6.6 it must be possible to regulate the speed at small values of flow with very great precision. The Table has been compiled with the help of additional pump curves and by interpolation.

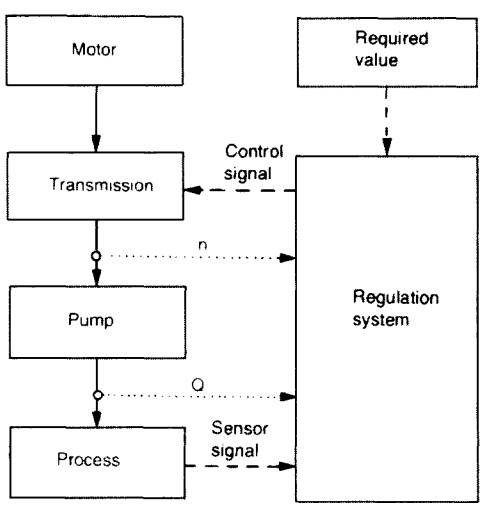

Figure 6.39 Schematic diagram for speed regulated pump

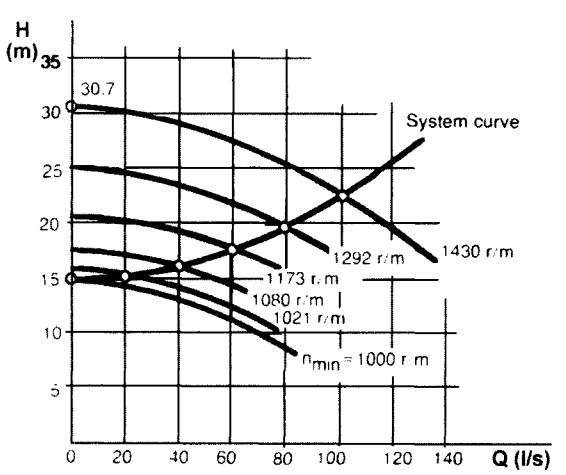

Figure 6.40 Examples of pump curves for determining the relationship between flow Q and speed n

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Ql/s	0	5	10	15	20	40	100
n r/min	1000.0	1001.3	1005.2	1011.7	1020.7	1080	1430

Table 6.6 Data compiled from Figure 6.40

Alternative methods are to linearise the speed by means of so-called indirect flow measurement or for example to eliminate speed completely from the control loop and to replace it by flow. When precise flow control is required, the method adopted for flow measurement must be selected with great care. It must be emphasised that rotodynamic pumps produce some cyclic flow variations and pressure pulsations; centrifugal pumps at high speed can be quite unacceptable. Flow measurement by orifice plates, for example, can be impaired by noise in the signal and may require averaging or smoothing.

Positive displacement pumps can suffer from complex relationships for speed and flow. Slip tends to be a variable rather than a constant. Changes in speed, differential pressure, and in some designs temperature, make it advisable to use flow measurement as the primary control signal. Metering pumps usually run slow enough for slip to be constant; direct speed control will produce proportional flow control.

The dynamic interaction between a speed regulated pump, reservoir or pump sump and pipeline can also be a cause of instability. In such a case the problem has to be studied with regard to the response time of the speed regulation, time constant, delay time, filling time of the reservoirs and transient changes in the flow requirement due to the opening or closing of valves, among other things. Relatively slow changes in pump speed are always preferred.

To build up an installation comprising the various component parts, without carrying out a stability analysis from the point of view of regulation can involve considerable risks. When buying packaged solutions one should, for the same reason, try to obtain functional guarantees based upon the actual case. The process parameters must then be clearly specified as regards pipeline characteristics, size of pumps, reservoirs, storage and operational conditions, including variations in the liquid properties.

Local packaged systems are available commercially as an alternative to the user selecting component parts in order to make up a complete system. These include pump, drive equipment with speed regulation and control system. In some cases several pumps are included, of which at least one is speed regulated. The control system then comprises facilities for the switching in and out as well as providing for interchange between the various pumps, so that the method of operation can be chosen to suit the total function of the process. An interface to plant control systems can be fitted by using one of the standard data communication interface protocols; such as 4-20 mA, RS 232, RS 423, IEEE 488, and 60488 or HART and others; depending upon the nature and complexity of the data.

6.9.8 Schematics for various systems

Schematics are an important tool in system design and control logic. A large amount of information about the various piped and electric/electronic systems can be communicated using simple symbols and boxes. The symbol used for a pump, in the examples shown in Figures 6.42 to 6.48, complies with several national standards.

This symbol should not be used in actual schematics because it does not convey anything about the pump type. If engineers of different disciplines view and use a schematic, recommendations and improvements can be suggested based on the pump types shown.

The symbols shown in Figure 6.41 are proposed for consideration.

6.9.8.1 Variable ratio coupling

A variable ratio coupling can be hydraulic or electric, see Figure 6.42. Fluid couplings and hydrostatic drives are torque devices, torque convertors are almost constant power devices and eddy current couplings are slightly better than constant torque. Fluid couplings, hydrostatic drives and torque convertors can be controlled electrically or pneumatically, eddy current couplings are usually controlled electrically.

Semi-automatic multi-speed gearboxes can be controlled in a similar manner. The gearbox would be "pulsed" by a digital signal, which triggers the pneumatics, to change up or down in response to the demand. Mechanical variators can usually be controlled electrically or pneumatically.

6.9.8.2 Voltage regulated induction motor

For an AC induction motor the torque is proportional to the square of the applied voltage. Speed regulation can be obtained by varying this. A limiting condition, however, is that the motor should be equipped with a rotor having increased resistance, a so-called high resistance rotor, in order for the losses to be reasonable. The variable voltage is obtained by means of a transformer or a thyristor connected in the motor supply, Figure 6.43. The minimum speed depends on the motor character-

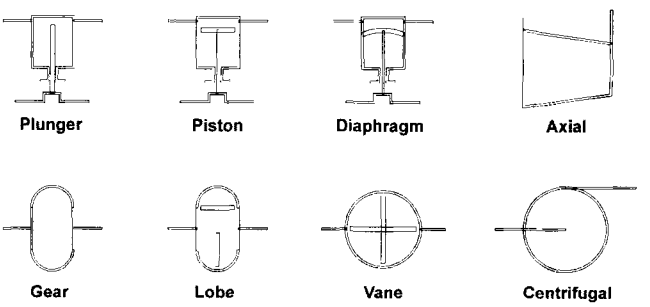

Figure 6.41 Typical symbols for different pump types

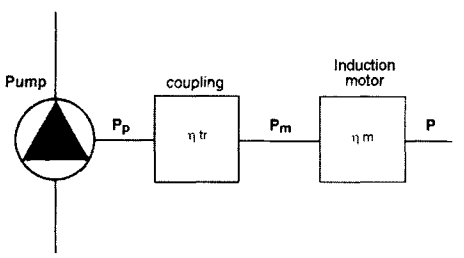

Figure 6.42 Schematic for speed regulation using variable ratio coupling

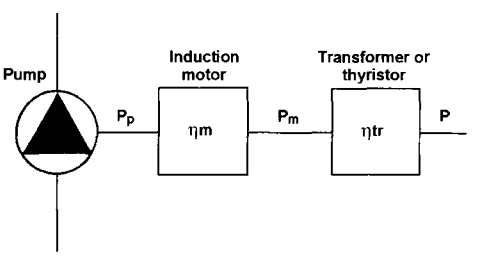

Figure 6.43 Schematic for voltage regulated AC motor

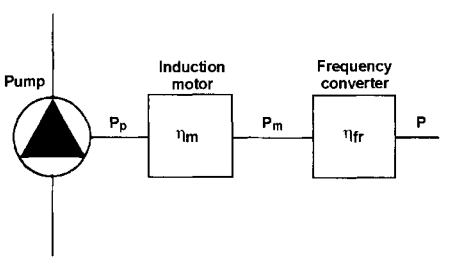

Figure 6.44 Schematic for frequency regulation

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istics, mainly the magnitude of the rotor resistance. Control would normally be electric.

6.9.8.3 Frequency regulated induction motor

The speed of an AC induction motor is directly proportional to the frequency. In frequency regulation the frequency and also the voltage to the motor is varied simultaneously. The frequency converter, or inverter, is connected in the motor supply, Figure 6.44. In the frequency converter, the mains power is first rectified and then the DC current is chopped up to form AC at the required frequency and voltage. Because the frequency converter is electronic, electric control is the ideal; all the protocols listed in Section 6.9.6 are used depending upon the control source.

6.9.8.4 DC motor drive

For a DC motor with constant field, the speed is approximately proportional to the armature voltage. The armature voltage can be controlled quite simply for any required value by using a thyristor rectifier to convert AC to DC, Figure 6.45. Electric control signals similar to frequency converters.

6.9.8.5 Slipring induction motor

The speed of an AC slipring or wound rotor induction motor can be controlled by taking power from the rotor via the sliprings, Figure 6.46. The power can be dissipated in an external regulating resistance or fed back into the mains via a thyristor rectifier. In the latter case the rotor voltage is rectified and then chopped back into AC and is fed back into the mains via a transformer or sub-synchronous cascade rectifier. Electric control similar to frequency converters is standard.

6.9.8.6 Engines

Industrialised engines are fitted with governors which control the speed. The governor is set to a specific speed and adjusts the fuel system to maintain the speed as the load varies. External speed control systems send signals to the engine governor to adjust the set speed. Most governors have a "tick-over" or "idling" setting which allows the engine to run slowly during warming-up or when the pump torque is low, on by-pass or unloaded. Governors are available to match any control signal. Figure 6.47 shows a typical engine arrangement.

6.9.8.7 Air motors

The speed of air motors is regulated by a flow control valve in the air supply. Control valves may be actuated by air, liquid or directly by an electric positioner. The control signal can be pilot air or liquid or digital or analogue electric. The pump flow should be measured and the speed adjusted to produce the correct flow, see Figure 6.48.

6.9.8.8 Turbines

Steam and gas turbines are fitted with governors in a similar manner to engines. The governor is set to the desired speed. In steam turbines, the governor actuates steam valves which control how much steam and how many nozzles are in operation to provide the power to operate at the set speed. Gas turbine control is identical to engines, the governor controlling the fuel supply, see Figure 6.47.

6.9.8.9 Turbo-expanders and hydraulic turbines

Both turbo-expanders and hydraulic turbines derive their power supply from flowing pressurised fluid streams. Neither device has any control over the fluid source so a flow control valve must be used to regulate the proportion used. Additional controls may be required in the primary supply to divert or by-pass fluid which is not used because of load conditions. The pump user and the manufacturer must agree who supplies the valves. In principle, control is exactly the same as for air motors, see Figure 6.48.

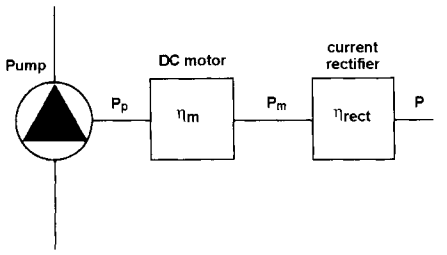

Figure 6.45 Schematic for DC motor drive

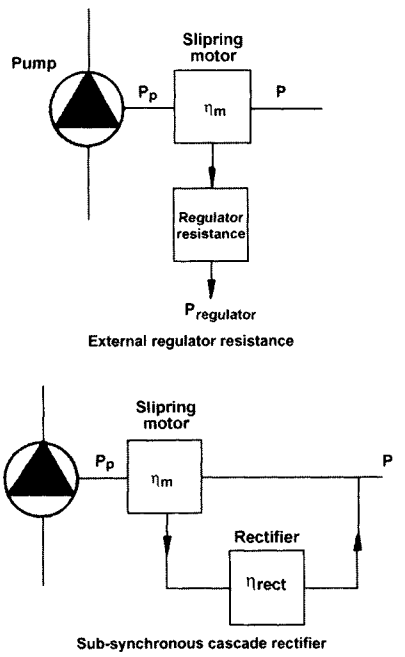

Figure 6.46 Schematic for speed regulation using a slipring motor

6.10 Choice of flow regulation method

6.10.1 General

The choice of flow regulation in pumping has traditionally been made largely on the basis of the initial investment required for the equipment. This method of assessment can be blamed on design/procure/build contractors who are not responsible for the ultimate running costs of the plant. As a result of increased energy costs, however, it has become essential to take into account the total costs during the whole life of the equipment, "life cycle cost", see also Chapter 14. This approach brings with it a drastic change of established practice, meaning among other

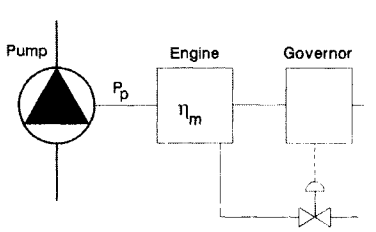

Figure 6.47 Schematic for engine speed regulation

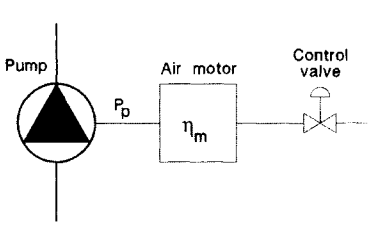

Figure 6.48 Schematic for air motor control

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things, that throttle regulation is often being replaced by speed regulation.

In order to make a correct choice of regulation method, it is necessary to carry out an extensive technical/economic analysis where consideration is given to:

- System requirements for cyclic flow variations and pressure pulsations and flow range
- Constancy of flow throughout the life of the equipment
- Shape of the system curve and the variation in the system curve due to start-up/shut-down, corrosion, fouling, seasonal production
- Criticality of service, how much of the process depends upon the operation of this unit
- Reliability, i.e. the general requirement for back-up and stand-by equipment
- The ability to modify the equipment slightly or significantly to cope with process changes due to future refinements or modifications
- Primary control system
- Sensitivity to environmental conditions, particularly humidity, dust, temperature range, gases such as methane, ammonia and hydrogen sulphide
- Serviceability, particularly with regard to the various skills required of local personnel involved, e.g. electrical, electronic and mechanical
- Energy costs of all available sources, including variations during the life of the equipment and/or seasonal variations
- Initial cost of equipment
- Life, cost and availability of spare parts
- Service life of the equipment and estimated profits

The flow/head required and constancy of flow decide whether the flow shall be distributed among several pump units, of which in its turn only one may need to be flow regulated.

The shape of the system curve and constancy of flow will guide the selection of pump type and flow regulation method.

The need for a stand-by unit, and if the stand-by units requires regulation, is dependent upon the criticality of the service. Reliability of the unit may be improved by imposing limitations on the normal operation; reduced speed, reduced power, reduced operating pressure. The requirement to have a stand-by unit affects the investment calculation significantly; additional space, piping, valves, cables, control equipment; since the cost savings from maintaining production have to cover the investment and operational costs of the stand-by unit.

As can be seen in Chapter 14, the cost of energy is the dominating operational cost, for which reason a partial optimization of investment for regulation equipment and energy savings will give satisfactory results. The energy costs can be determined by use of the total efficiency factor, see Section 14.3.1, or by use of total power consumption at various flows.

6.10.2 Direct flow regulation

Table 6.7 compares the power requirements for flow control of a centrifugal pump using different speed methods and with flow regulation using throttle and on-off regulation to the hydraulic power.

	Flow				
	20	40	60	80	100
Hydraulic power kW	0.8	6	22	51	100
Variable frequency AC (large)	1	6.5	24	55	109
motor or DC motor (small)	1.5	8.5	31	72	140
Mechanical variators	8	16	33	61	110
Fluid coupling	10	23	44	74	115
Torque convertor	8	18	37	61	125
Hydraulic coupling	10	19	38	72	129
Throttling	84	90	97	106	110
On-off control (see Figure 6.6)	24	51	76	97	110

Table 6.7 Comparison of power requirement with various methods of flow regulation of a centrifugal pump with zero static differential head

The comparison applies to one specific case in which the system's static differential head is zero, i.e. the only losses are pipe frictional ones. For other operational cases, particularly with increasing proportion of static differential head, the power savings are less for speed regulation. When the differential head consists only of static head, there is in certain cases a 10% to 20% greater power consumption for speed regulation than for on-off control. In these circumstances a positive displacement pump may give better results, see Figure 6.38.

The loss of head in the control valve in accordance with equation 6.17 is not included. This corresponds in this case to a power of about 35 kW.

From Table 6.7 it can easily be seen that throttling is inefficient. Throttling, or by-passing, should only be undertaken when the flow reduction necessary is small and/or the operating time at reduced capacity is short. On-off control is inefficient for flow regulation on purely friction systems. If a cheap reliable source of energy can be obtained the cost of inefficiency can be negated and the choice of method will be based on initial installation costs.

6.10.3 Speed regulation

To determine energy consumption it is necessary to summate power consumption over a period of time using a constancy diagram. If there is no constancy diagram, a rough estimate can be made using mean flow, or better still that flow which has 50% constancy. When studying the constancy of flow together with the performance of the pump already chosen, it is usually difficult to differentiate between the extreme flow requirement and the approximate safety margin. The normal maximum flow is generally less than 80% of the pump flow. The flow with 50% constancy is therefore usually quite low and can be judged to be:

- About 60% of maximum pump flow in process industries
- About 30% of maximum pump flow in public water supplies
- About 40% of maximum pump flow in heat transport (central heating and district heating)

Reducing safety margins and operating pumps closer to their BEP will lower operating costs.

A comparison of the power requirements in Table 6.7 at 60% of mean flow shows a difference between throttle regulation and speed regulation by fluid coupling of 97- 44= 53 kW. For example, for a capitalised cost for 1 kW of about $525, the difference in the capitalised energy costs will be about $28,000. Since the additional costs for speed regulation by fluid couplings could be about $7,000, the economic superiority of speed regulation is indisputable, see Figure 6.27.

This example illustrates that in many cases speed regulation is well worth considering. The cost of energy is critical. If energy is virtually free, e.g. waste steam or high pressure gas/liquid, there is no need to do any calculations. The energy source must be available for the lifetime of the pump though. As a result of general studies, it can be said that speed regulation of rotodynamic pumps for a continuous flow requirement is the best method of regulation at virtually all powers, if the system

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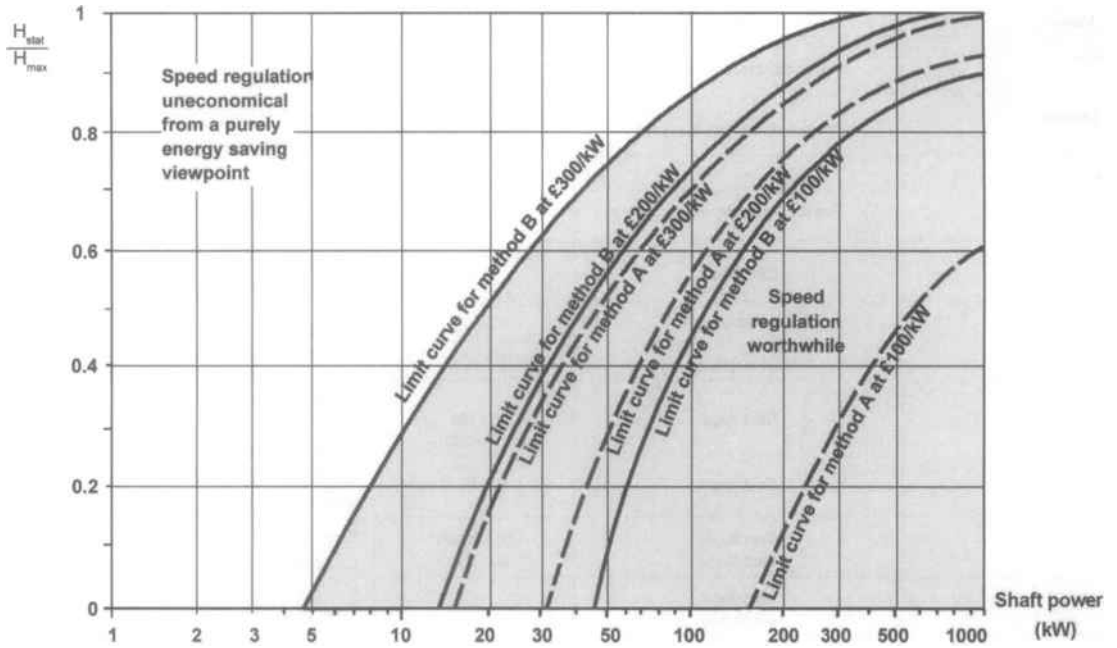

Figure 6.49 Approximate regions of viability for speed regulation. Method A applies to AC frequency regulation and B to fluid couplings
Figures in £/kW comprise capitalised energy costs based on maximum shaft power, see Table 14.8, in Chapter 14

curve comprises pipeline losses only. A review of other cases for rotodynamic pumps is shown in Figure 6.49. It can be seen that the capitalised value of energy costs per kW and the shape of the system curve, H_{stat}/H_{max} as in Figure 6.50, has considerable effect.

The limit of economic viability for speed regulation according to Figure 6.49 gives a first guide regarding the choice of method of regulation for rotodynamic pumps. As H_{stat}/H_{max} increases, positive displacement pumps should be considered. As a second step in the assessment, the investment costs for the actual installation can be compared with the possible energy savings using speed regulation. Intensive studies of normal pump installations actually show that possible energy savings using speed regulation compared to throttle regulation amount to 20% to 60% and depend heavily upon the shapes of the system and pump curves.

The guideline values shown in Figure 6.51 for energy savings can serve as a guide at the initial project stage and apply most closely to the process industries. In this illustration, Method A includes mechanical variators, frequency controlled AC motors and thyristor controlled DC motors; Method B includes fluid couplings and eddy current couplings. For water mains, central heating and domestic purposes, the energy savings are greater because of low mean flow. When the flow constancy is known, it is then of course possible to determine energy consumption for various regulation methods for the actual pump installation. The calculations can be simplified to a great extent by using Simpson's Rule as shown in previous Sections.

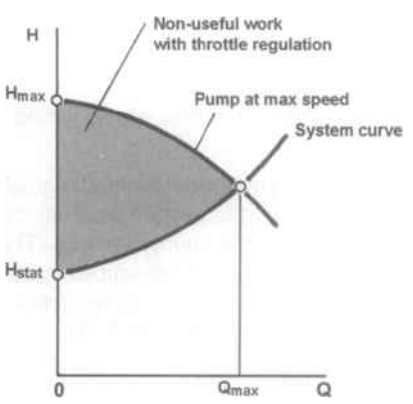

Figure 6.50 The system curve and centrifugal pump curve and their effect on method of flow regulation determined by the parameter H_{stat}/H_{max}

Safety factors applied to flow and differential heads carry with them dramatic increases in costs of pump installations. A flow safety margin of just 25% can double both the investment and the operational costs for a pump installation. The consequence of the increased power consumption due to the safety margin is eased to some extent with speed regulation as can be seen in Table 6.8, a summary of costs per kW of shaft power:

	Throttle reg £/kW	Speed reg £/kW
Increase in pumps costs	15-60	15-60
Increase in motor costs	20	20
Increase in costs for reg equipment, approx.	10	50
Capitalised energy costs, approx.	300	50
Total of additional costs, approx.	345/390	135/180

Table 6.8 Summary of costs per kW of shaft power

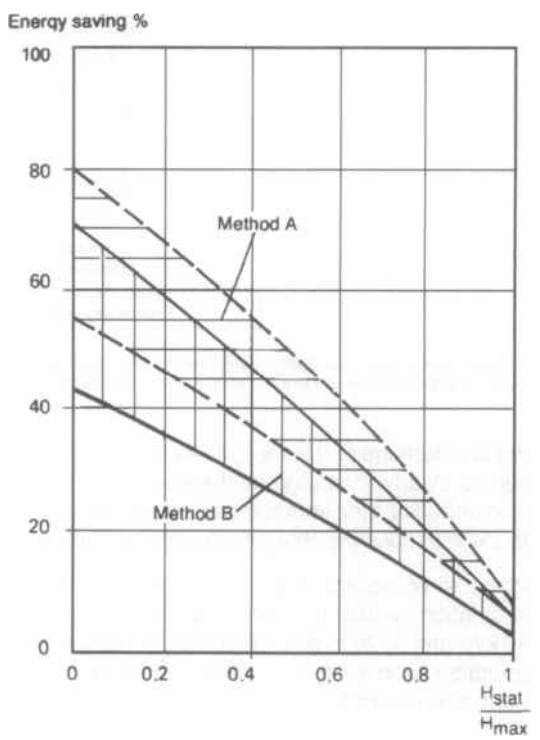

Figure 6.51 Guide-line values of normal energy savings in the process industries with speed regulation of centrifugal pumps relative to throttle regulation

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Method	Pumps	Power kW	Application
Multi-speed gearbox	Positive displacement pumps	10 - 2000	Pipeline pumps Transfer pumps
Semi-automatic gearbox	Positive displacement pumps	100 - 400	Engine driven portable test pumps Oil-well fracturing
Mechanical variator	Metering pumps Positive displacement pumps	1 - 10 10 - 200	Chemical dosing General purpose Variator can include speed reduction
Hydrostatic transmission	Centrifugal	5 - 100	Ship bilge pumps One power pack drives several remote motors
Hydrostatic coupling	Centrifugal Positive displacement	5 - 30 5 - 30	Drive prevents over-running and turbining Constant torque drive for high H_{stat}
Fluid coupling	Centrifugal Centrifugal	5 - 100 10 - 200 50 - 5000	Sewage pumps Liquor & pulp pumps Hostile environment Circulating pumps for district heating Very reliable
Torque converter	Centrifugal Centrifugal	50 - 20000 50 - 5000	Boiler feed pumps Circulating pumps
Eddy current coupling (*)	Centrifugal Mixed flow	5 - 300	Water works & sewage
Multi-pole AC motor	Centrifugal Mixed flow Axial	0.05 - 25	Short pipeline pumps Transfer pumps Circulating pumps
Frequency controlled AC motor (*)	All pump types	0.1 & up	Most applications
Variable speed Switched reluctance motor	Centrifugal Positive displacement (rotary) Positive displacement (reciprocating)	up to 300 up to 150 up to 300	High speed pumps General process and hydraulic power packs
Primary voltage controlled AC motor	Centrifugal Mixed flow Axial	0.01 - 20	Short pipeline pumps Circulating pumps Booster pumps
Slipring AC motor	All pump types	20 - 500	Water works Sewage
DC motor (*)	All pump types	1 - 1000	Applications requiring precise speed control and rapid response
Petrol engine Diesel engine	Centrifugal Mixed flow Positive displacement	1 - 300	Portable pumps for site work, drainage Portable hydrotest pumps
Gas engine Crude oil engine	Centrifugal Positive displacement	100 - 3000	Pipeline pumps
Air motor	Centrifugal Mixed flow Axial	1 - 25	High speed pumps up to 9600 r/min from std compressed air
Steam turbine	Centrifugal Mixed flow Axial	25 & up	Boiler feed pumps Used in refineries as main pump, electric stand-by
Gas turbine	Centrifugal Mixed flow Axial	25 & up	Used where steam & electricity not available, popular off-shore
Turbo-expander	Centrifugal Mixed flow Axial	10 & up	Used in continuous processing, refineries with hp gas streams
Hydraulic turbine	Centrifugal Mixed flow Axial Positive displacement	10 & up	Used in continuous processing, refineries, RO, with hp liquid stream

Table 6.9 Typical applications for speed regulation RO ≡ reverse osmosis fresh water production
(*) Control equipment requires cooling air free from excess humidity

The need to determine real pump data as early as the initial project stage, is clearly illustrated by the above costing example which also underlines the importance of modifying pump data in accordance with possible changes in operational conditions.

The initial cost of the actual pump equipment increases with speed regulation by 80 to 150 £/kW shaft power in small pumps, about 10 kW, and by 20 to 80 £/kW for larger pumps, about 100 kW, depending upon the type of speed converter and ancillary equipment, see Figure 6.27.

All of the previously mentioned types of drive equipment are suited to various types of application. Table 6.9, which is by no means exhaustive, shows just how widely varied applications and sizes are.

When comparing various methods of speed regulation it should be noted that it is the speed regulation itself which is responsible for the greater part of the energy savings. The method of speed regulation and its transmission efficiency only have marginal effects, as can be seen from the following examples of typical investment calculations, see Table 6.10.

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	DC motor drive AC variable frequency	Fluid coupling Eddy current
Capital investment costs	£15000	£8000
Savings in energy costs £ p.a.	£5000	£4000
Pay back time	3 years	2 years

Marginal investment	£15000 - £8000 = £7000
Marginal savings	£5000 - £4000 = £1000 p.a.
Pay back time	7 years

Table 6.10 Typical investment calculations

On the basis of these calculations both groups seem to be entirely acceptable. If the marginal investment is now studied, we get: The pay back time of 7 years is matched roughly by the write-off time of 30 years at the interest factor 15% and means that variable speed motors in this case are uneconomical compared with variable speed couplings.

In the same way as in the marginal calculation one may find that variable speed motors offer the same economic viability as variable speed couplings only when the procurement costs for motors do not exceed couplings by more than 20% to 30% i.e. the same value as corresponds to the greater energy savings of motors.

6.11 Useful references

HART Communication Foundation, 9390 Research Boulevard, Suite I-350, Austin TX 78759 USA. Tel: 512-794-0369, Fax: 512-794-3904, www.hartcomm.org.

IEEE488.2 -1992 (R2004) IEEE Standard Codes, Formats, Protocols, and Common Commands for Use with IEEE Standard. 488.1-1987, IEEE Standard Digital Interface for Programmable Instrumentation, (Now replaced by IEC/IEEE 60488 -2-2004).

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