In order to minimise head losses and, in particular to avoid an excessively high wear rate of the conical enlargement and the adjacent pipework, it is good design practice to adopt the included angle, \( \theta = 6^\circ \). In practical terms, this is equivalent to allowing an increase in diameter of 10mm for every 100mm in length of the conical section.

**FIGURE 3-2 TYPICAL CONICAL ENLARGEMENT**

- **Pump Feed Hoppers**
  Consideration should be given to the design of a suitable pump feed tank or hopper. Some basic guidelines are given in Section 2.8(b).

- **Shaft Sealing**
  It is important that the correct type of shaft seal is selected to suit the specific duty conditions. Further details on the various types of seals available can be found in Section 2.8(a).

- **Multi-Staging**
  Where the duty required exceeds the head limitations of a single pump, multiple pumps in series may be required. Further details on the considerations required for series pumping can be found in Appendix 7.

- **Drive Selection**
  Direct coupling to fixed speed motors is common with non-slurry type centrifugal pumps. Duty variations are usually achieved through variations in impeller diameter.

For slurry pumps, impellers are constructed in hard metal alloys or metal-reinforced elastomers. It is usually not economical or practical to reduce the diameter of standard impellers to meet specific duty requirements.

Throttle valves are not recommended for use in slurry systems to control flow rate (by head loss), due to the resultant higher values of head, speed, wear on pump and valves, and due to the increased power required. An additional restriction is the increased risk of pipeline blockages.

Slurry pump impellers of standard diameter are recommended for most abrasive slurry pump applications. The optimum pump speed, or speed range, must be achieved by suitable means, (for example, vee-belt drive or variable speed drives).

Progressive Speed Changes are beneficial, by providing the lowest practical pump speed to handle the required duty at any given time.

The lowest speed results in the lowest power consumption and lowest wear rate, for any given Warman slurry pump applied to a given duty.

For duties up to approximately 250 kW, vee-belt drives from a fixed-speed motor, are commonly employed (although belt drives are available for up to 700 kW typically). The pump speed may be changed, as required, by occasional changes in vee-pulley ratio, (for example, a larger diameter motor pulley may be fitted). Where frequent variations are required, this solution is unattractive because the pump must be stopped to change speed, and the speed changes are stepped.

On some duties, the required pump speed may have to be varied progressively, possibly over a relatively wide range,

- a. due to wear,
- b. in order to maintain the intake static head at a constant value, or
- c. due to variations in required flow rate, static head, pipeline length exit pressure head or solids concentration.

Typical examples are:

- a. tailings disposal,
- b. mill classifier (cyclone) feed in closed-circuit grinding operations, and
- c. some variable suction dredging duties.

The most effective means of satisfying these progressively varying requirements is to provide for an efficient variable speed drive.

Where a motor size exceeds the practical limitations for vee-belt applications, a direct coupled motor in conjunction with a speed reducing gear box may provide the most practical solution.

### 3.12 TYPICAL PUMP CALCULATION

A heavy duty slurry pump is required for the following duty:

- 65 tonnes per hour of sand
- Specific gravity of solids 5 = 2.65
- Average particle size d50 = 211 microns (0.211mm)
- Concentration of solids Cw = 30% by weight
- Static discharge head (Zd) = 20 metres
- Suction head (Zs) = 1 metre (positive)
- Length of pipeline = 100 metres
- Valves and fittings = 5 x 90° long radius bends

The pump will be gravity fed from a hopper and be arranged generally as shown in Figure 3-3.

The pump size, speed, shaft power and recommended size of delivery pipeline are determined as follows:
a. The quantity to be pumped can be determined thus:

- Weight of solids in slurry = 65 tonnes
- Weight of volume of water equal to solids volume = \( \frac{65}{2.65} = 24.5 \) tonnes
- Weight of water in slurry of \( \text{Cw} = 30\% \) = \( \frac{65 \times (100-30)}{30} = 151.7 \) tonnes
- Total weight of equal volume of water = 151.7 + 24.5 = 176.2 tonnes
- \( * (1 \text{M}^3 \text{of H}_2\text{O} = 1 \text{tonne}) \)
- Total weight of slurry mixture = 65 + 151.7 = 216.7 tonnes
- Specific gravity of slurry mixture [sm] = \( \frac{216.7}{176.2} = 1.23 \)
- Concentration of solids by volume (Cv) = \( \frac{100}{176.2} \times 24.5 = 13.9\% \)
- Quantity of slurry = 176.2 m³/hr
  \(- = 49 \text{ litres/second} \)

b. Size of pipeline

A 150mm pipeline is selected as being potentially suitable and is checked as follows:

The slurry mixture volume is determined by the following formula:

\[ V = \frac{Q \times 1273}{d^2} \]

where
\[ V = \text{slurry velocity in metres/second} \]
\[ Q = \text{slurry flowrate in litres/second} \]
\[ d = \text{pipe diameter in mm} \]
\[ g = 9.81 \text{ m/s}^2 \]

Velocity \( V \) in this case is therefore:
\[ \frac{49 \times 1273}{150^2} = 2.8 \text{ metres/second} \]

Using Durand's equation from Appendix 5

\[ VL = \frac{FL \sqrt{2gD}}{3.14} \]

where
\[ D = \text{Pipe diameter in metres} \]
\[ L = \text{Pipe length} \]
\[ F = \text{Friction factor} \]
\[ V = \text{Velocity} \]
\[ L = \text{Length} \]
\[ D = \text{Diameter} \]
\[ g = \text{Gravity acceleration} \]
\[ 
\]

The value of \( FL \) is obtained from Figure 4-5-2, using a \( CV \) of 13.9\% and an average particle size \( d_{50} = 211 \) microns. (For widely graded particles).

Value of \( FL \) = 1.04

By substitution of values in Durand's equation the limiting settling velocity \( VL \) becomes:

\[ VL = 1.04 \sqrt{2 \times 9.81 \times 0.15 \times (2.65 - 1)} \]
\[ = 2.3 \text{ metres per second}. \]

The 150mm pipe is therefore considered suitable for this application since the limiting settling velocity (2.3 m/s) is lower than the actual slurry mixture velocity (2.8 m/s).

c. Friction head \( Hf \) for the pipeline

Firstly determine the equivalent length of pipeline, using the valves and fittings head losses table as shown in Figures 4-4-3 and 4-4-4.

- Actual length of line = 100 metres
- 5 x 90° long radius bends at 3.35 metres each = 16.75 metres
- Equivalent length of line = 116.75 metres

Using the steel pipeline size of 150mm and a slurry mixture velocity of 2.8 metres per second, the value \( f = 0.017 \) is obtained from Figure 4-3-2.

By substitution in Darcy's equation for friction head as in Figure 4-3-2:

\[ Hf = 0.017 \times \frac{116.75 \times 2.8}{0.15 \times 2 \times 9.81} \]
\[ = 5.29 \text{ metres of mixture for 116.75 metres of pipe}. \]

d. Loss in discharge pipe enlargement

It is also likely that a divergent pipe section will be required in the discharge pipe as a preliminary review of pump selections (Figure 3-1) indicates a pump with a 100mm diameter discharge to be a likely selection. A pipe transition piece would be required in this case to enlarge the discharge to the 150mm pipeline size.

This is dealt with in Figure 4-4-4. Head loss in this case using an enlargement included angle of 30° would be:

\[ \frac{Ke (V^2 - V_1^2)}{2g} \]
\[ = 0.41 \text{ metres} \]

e. Loss at pipe discharge

Under normal open discharge conditions, the velocity head at the pump discharge must be added to the required total head.

In this case the velocity head \( \frac{V_1^2}{2g} \) is \[ \frac{2.8^2}{2 \times 9.81} = 0.4 \text{ metres of mixture}. \]

f. Loss of head at entrance to suction pipe

This is dealt with in Figure 4-4-4. The suction pipe in this case is most likely to be similar to the discharge (150mm). Assuming the hopper would be fitted with a flush type connection, the appropriate loss would be:

\[ 0.5 \times \frac{V_2^2}{2g} = 0.2 \text{ metres} \]

\[ = 2g \]

g. Total dynamic head on the pump. [Refer Figures 4-4-1 and 4-4-2]

\[ Hm = \Delta Z + Hf \]

where \( Z \) is static head; ie: \( [Zd - Zs] \)

\[ Hm = (20 - 1) + 5.29 + 0.41 + 0.4 + 0.2 \]
\[ = 25.4 \text{ metres of slurry mixture}. \]

h. Equivalent water total dynamic head

From Figure 2-3, we are able to determine that the
appropriate correction (Hr/ER) in this case is 0.89.

The total head of equivalent water (Hw) is therefore:

\[ \text{Hm} = \frac{25.4}{0.89} = 28.53 \text{ m} \]

Say, 28.5 metres of water

i. **Pump selection**

The pump can now be selected, using the required flow rate of 49 litres/second.

Total head of 28.5 metres of equivalent water and a slurry SG of 1.23.

In this case, a Warman 6/4 D-AH heavy duty rubber lined pump is selected with a 5 vane closed rubber impeller at a pump speed of 1130 rpm (from Figure 3-4).

The consumed power at the pump shaft can be computed using a pump efficiency of 66% (from Figure 3.4) thus:

\[ \text{Hm} \times \text{Sm} \times 1.02 \times \text{em} \]

or

\[ \frac{\text{O} \times \text{Hw} \times \text{Sm}}{1.02 \times \text{ew}} \]

\[ = \frac{49 \times 28.5 \times 1.23}{1.02 \times 66} = 25.5 \text{ kW} \]

in this case, a 30 kW drive motor would be selected.
APPENDIX 4

c. Calculations

Cross-sectional area of Pipe at inside Diameter
\[ = \frac{D^2 \pi}{4} \times 3.1416 = 0.031416 \text{ m}^2 \]
As 1 m\(^3\) = 1000 litres,
Flow rate in m\(^3\)/s \[ = \frac{Q}{1000} = \frac{94.25}{1000} = 0.09425 \text{ m}^3/\text{s} \]
As \( V = \frac{\text{Flow rate (m}^3/\text{s})}{\text{Cross-sectional area of pipe (m}^2\text{)}} \)
Average Pipeline Velocity \( V = \frac{0.09425}{0.031416} = 3.0 \text{ m/s} \)
Alternatively,
\( V = \frac{1273 \times 0.09425}{200 \times 200} = 3.0 \text{ m/s} \)

Refer to the Warman Pipe Friction Chart, Figure 4-3-2.

As illustrated with arrowed lines, the chart is entered at the right hand bottom scale, along the applicable ‘d’ co-ordinate and, at its intersection with the appropriate (pipe surface material) reference line, the corresponding ‘k/d’ co-ordinate is followed across, towards the left hand portion of the chart, until it intersects the “NR = 10” co-ordinate. From this intersection, the ‘k/d’ co-ordinate is drawn as a curve following the geometry of the adjacent family of curves.

(* Reynolds number (NR) is an expression for the ratio of inertia forces to viscous forces.)*

The left hand portion of the chart is entered separately via a line drawn across the nomogram axes ‘d’ and ‘V’, at their applicable values, to intersect the ‘f = 0.008’ co-ordinate. This is equivalent to entering the chart at the appropriate value of Reynolds Number (NR) for clear water at 20°C. From this intersection, the ‘NR’ co-ordinate is followed until its point of intersection with the ‘k/d’ curve which has been drawn in the previous step.

This point lies on the required ‘f’ value co-ordinate:

\( f = 0.0158 \)

Thus the value of friction loss, \( H_f \) can be evaluated as follows:

\[ H_f = \frac{f \times \frac{V^2}{2g}}{D} \]

\[ H_f = 0.0158 \times \frac{700}{0.200} \times \frac{3.0^2}{2 \times 9.81} = 25.4 \text{ m} \]

TOTAL DYNAMIC HEAD

4.1 ABSTRACT

The main components of Total Dynamic Head are:

a. Total Discharge Head, and
b. Total Suction Head.

The equation is,

\[ \text{Total Dynamic Head} = \text{Total Discharge Head} - \text{Total Suction Head} \]

Algebraically, \( H = (\Delta H_d - (\Delta H_s)) \) (m)

or, \( H = (\Delta H_{gd} + H_{vd}) - (\Delta H_{gs} + H_{vs}) \) (m)

The values \( H_{vd} \) and \( H_{sv} \) are always positive (+ve)

\( \Delta H_d \) is usually positive (+ve), (above pump centreline)

\( \Delta H_s \) may be positive (+ve), (above pump centreline) or

negative (-ve), (below pump centreline).

When \( \Delta H_s \) is positive (+ve):
\[ H = (\Delta H_d) - (\Delta H_s) \] i.e. \( H = H_d - H_s \)

When \( \Delta H_s \) is negative (-ve):
\[ H = (\Delta H_d) - (\Delta H_s) \] i.e. \( H = H_d + H_s \)

a. Total Discharge Head, \( H_d \)

Basic Simple Formula: \( H_d = Zd + H_{vd} + H_{ve} \) (m)

\( Zd \) may be positive (+ve) or negative (-ve)

If applicable, additional terms must be included in the formula to account for increased value of \( H_d \), due to any contractions (for example, nozzle friction loss) and enlargements; friction loss in a flow-measuring device and exit into pressure-fed equipment, for example, a hydraulic cyclone.

b. Total Suction Head, \( H_s \)

Basic Simple Formula: \( (\Delta H_s) = (\Delta H_i) - Zs - H_{sv} \) (m)

\( \Delta H_i \) and \( Zs \) may each be positive, (+ve) or negative, (-ve).

If applicable, additional or substitute terms must be included in the formula to account for increased or decreased values of \( H_s \) due to any contractions, enlargements, flow measuring device. These are as follows:

i. liquid supply surface being under pressure, \( H_{pr} \), or under vacuum, \( H_{vac} \),

ii. differential column head loss, \( Z_c \), and

iii. substitution of effective mixture static suction head \( Z_{sm} \) in lieu of \( Z_s \).

NOTE: Values of \( H_s \) are directly applicable in NPSHa calculations and in selection of shaft-sealing arrangements.

4.2 RELATIONSHIPS BETWEEN HEAD, SPECIFIC GRAVITY, AND PRESSURE OR VACUUM

The term “Total Dynamic Head” correctly describes the driving force developed by a centrifugal pump, regardless of the Specific Gravity of the liquid or slurry pumped. The head (+ve) or (-ve) at any point in the system may be
converted to pressure or vacuum, respectively, by the application of conversion formulae.

### 4.3 TOTAL DYNAMIC HEAD

Total Dynamic Head, \( H \), is the head which is required by a given system to maintain a given flow rate, \( Q \), through the system.

\( H \) varies as the flow rate through the system, \( Q \), varies. The relationship of \( H \) with \( Q \) is known as the system resistance and may be expressed algebraically or graphically.

#### a. Total Dynamic Head: With Positive (+ve) Suction Head:

Figure 4.4-1 illustrates a pump discharging a flow rate, \( Q \), with discharge and suction gauge pressure heads, both relative to atmosphere and both corrected to pump centreline, measured at the pumps discharge flange and at the pump suction flange, respectively. All heads are expressed in metres of actual mixture being pumped.

The Total Dynamic Head, \( H \), required to maintain flow rate \( Q \) through the system is the algebraic difference between the Total Discharge Head and the Total Suction Head.

\[
H = H_d - (H_s)
\]

\[
H_d = (H_{gd} + H_{vd}) - (H_{gs} + H_{vs})
\]

where \( H_{vd} = \frac{V_{d}^2}{2g} \) and \( H_{vs} = \frac{V_{s}^2}{2g} \)

#### b. Total Dynamic Head: With Negative (-ve) Suction Head:

When \( H_s \) is negative (-ve), that is, a vacuum head is indicated by the gauge, as in Figure 4.4-2, the substitution of the negative value in the formula serves to positively increase the value of \( H \) with respect to \( H_d \).

### 4.4 ESTIMATION OF TOTAL DYNAMIC HEAD

As \( H = (H_d) - (H_s) \) and as the suction and discharge pipes are often of different internal diameter, it is advisable to estimate values of \( H_d \) and \( H_s \) separately. The formulae used should be the Basic Simple Formulae, but amended where necessary to allow for any additional or substitute terms specific to the proposed duty as follows:
a. Total Discharge Head: \( H_d \)

Basic Simple Formula: \( H_d = (Z_d) + H_{fd} + H_{ve} \)

Typical Possible Additional Terms are as follows:

i. Head Loss on conical enlargement, (see Figure 4.4-4)

ii. Head Loss on contraction, (see Figure 4.4-4)

iii. Head Loss on Exit into Pressure-Fed Equipment, (refer to Section 2.8(d))

b. Total Suction Head : \( H_s \)

Basic Simple Formula: \( H_s = (Z_s) + H_l + H_{fs} \)

Typical Possible Additional Terms are as follows:

i. Head GAIN in supply from Pressure Vessel, \( H_{pr} \)

ii. Head Loss in supply from Vacuum Vessel, \( H_{vac} \)

iii. Head Loss in Differential Column. (Applicable in dredge applications), \( Z_c \) (refer A.5(e))

4.5 SEPARATE ESTIMATES OF SUCTION HEAD AND DISCHARGE HEAD

a. Pipeline Friction Head Loss, \( H_f \)

The friction head loss in a given pipeline is estimated for the Total Equivalent Length of Pipe, \( L \) (m), which is the sum of the Total Actual Length of Pipe, \( L_a \) (m) and the Aggregate of Equivalent Lengths for all valves, bends and like fittings, \( L_f \) (m), (see Figure 4.4-3) contributing to friction head loss in the pipeline.

Generally \( L = L_a + L_f \),

Specifically:

i. for Suction Side: \( L_s = L_{as} + L_{fs} \) (Friction Head Loss = \( H_{fs} \))

ii. for Discharge Side: \( L_d = L_{ad} + L_{fd} \) (Friction Head Loss = \( H_{fd} \))

Values of \( H_{fs} \) and \( H_{fd} \) should be estimated separately, for example, during the preparation of the respective separate sets of calculations leading to the estimates of \( H_s \) and \( H_d \). By separately estimating \( H_s \), its value is readily available for use in \( NPSH_a \) CALCULATIONS, (refer to Appendix 6), and in the selection of Shaft-Scaling arrangements, (refer to 2.8(a)).

b. Inlet Head Loss, \( H_i \): Exit Velocity Head Loss, \( H_{ve} \)

Separate provision is always made in the standard formulae for the terms:

i. \( H_i \), the Inlet Head Loss (Suction side only), and

ii. \( H_{ve} \), the Exit Velocity Head Loss (Discharge side only).

That is, the terms \( H_i \) and \( H_{ve} \) are included in the standard formulae for \( H_s \) and \( H_d \) respectively.

c. Head Losses due to Constrictions and

Enlargements

These additional head losses are calculated by use of the formulae provided in Figure 4.4-4. As no separate provisions are made in the standard \( H_s \) and \( H_d \) formulae for individual symbols or terms anticipating these friction head losses, any such estimated head losses, if applicable, should properly be added to the values calculated for \( H_s \) or \( H_d \) respectively.

Friction losses in jet nozzles \( [Hn] \) may be treated as for conical contractions unless more reliable head loss, data is available.

d. Sundry Additional Causes of Effects on \( H_s \) or \( H_d \)

The calculated values for \( H_s \) and \( H_d \) must be corrected to allow for permanent friction head losses when any in-line restrictions, such as flow-measuring devices, are intended to be installed (for example, quarter-circle orifice plates).

e. Differential Column Head Loss

Figure 4.4-5 depicts a mixture of Specific Gravity, \( S_m \), flowing upwards and drawn from a supply of settled solids and overlying liquid, \( S_l \). As the \( S_m \) is greater than \( S_l \), the vertical height \( Z_l \), of mixture in the submerged portion of the suction pipe is not completely balanced by the surrounding liquid of the same vertical height, \( Z_l \). The resulting effective static head loss is known as the Differential Column Head loss, \( Z_c \):

\[
Z_c = Z_l \left( \frac{S_m - S_l}{S_m}\right)
\]

Where this condition exists, \( Z_c \) must be included as an additional head loss in the pipe system. This would effect both total head and \( NPSH_a \) (refer to Appendix 6).

![FIGURE 4.4-5 DIFFERENTIAL COLUMN HEAD LOSS](image-url)
FIGURE 4-4-3  EQUIVALENT LENGTHS OF PIPE FITTINGS AND VALVES

GROUPS 1 105 IN TABLE SHOW THE APPROXIMATE PROPORTIONS OF VELOCITY HEAD, \( H_v = \frac{V^2}{2g} \), WHICH APPLY TO CERTAIN CONDITIONS.

\( V \) IS USED TO INDICATE THE UP STREAM VELOCITY AND \( V \) THE DOWN STREAM VELOCITY.

\[ H_v = \frac{V^2}{2g} \]

**FIGURE 4-4-4  HEAD LOSSES AT INLET, CONTRACTION AND ENLARGEMENT**
LIMITING SETTLING VELOCITY

5.1

Slurries containing essentially fine particles (predominantly less than 50 microns) are generally considered non-settling (homogeneous) and can normally be assessed without consideration for settling. In high concentrations however, these slurries often exhibit non-Newtonian flow properties (or rheology) and require special consideration in determining suitable pump and system parameters. Further information can be obtained by contacting your nearest Warman office.

Slurries containing particles predominately greater than 50 microns are generally considered settling (heterogeneous), which is the case in the majority of slurry pumping applications.

Slurries containing solid particles essentially coarser than 50 microns are transported in suspension by a liquid in a pipe, providing the average velocity, \( V \) is no less than the limiting settling velocity, \( V_L \). At any velocity below \( V_L \), solids are deposited in the pipeline. This results in increased pipeline friction head loss, with reducing flow rate and may lead to a blockage of the pipeline.

5.2 DETERMINATION OF LIMITING SETTLING VELOCITY

In order to determine \( V_L \) accurately, it is necessary to conduct tests with the slurry on a pipeline test rig. As a practical alternative, where this is not possible, the \( V_L \) may be established by a skilled specialist or estimated by one of the following methods, each based on Durand's formula:

\[
V_L = FL \sqrt{\frac{2gD}{(S - S_i)/S_i}}
\]

Where the parameter \( FL \) is dependent upon particle sizing and solids concentration.

Durand’s formula was derived initially from tests carried out on slurries of closely-graded particle sizing, (see Figure 4-5-1).

A closely-graded particle sizing, (for the purposes in this Handbook), is regarded as one where the ratio of particle sizes, expressed as testing screen apertures, does not exceed approximately 2:1, for at least 90% by weight of the total solids in the sample. Subsequent tests indicate that values of \( FL \) (from Figure 4-5-1), provide conservative (high) values for \( V_L \) in respect of:

a. Slurries of more widely-graded particle sizing, and/or

b. Slurries of sizing containing significant proportions of particles finer than 100 \( \mu \)m.

It is important that values of \( FL \) (and \( V_L \)) are not excessively conservative (high). Excessively conservative estimates of \( FL \) (and \( V_L \)) will result in the high pipeline velocities, high power consumption and high rates of wear on pipes and pumps.

Method (A) : ESTIMATING FL : CLOSELY-GRADED PARTICLE SIZING:

Given values for \( d_{50} \) and \( C_v \); values of \( FL \) are obtained from Figure 4-5-1.

Method (B) : ESTIMATING FL : WIDELY-GRADED PARTICLE SIZING:

Widely-graded sizing are more commonly encountered in slurry pumping operations.

Figure 4-5-2 represents the results of field tests on slurries of widely-graded sizing. The particle sizing is simply expressed by the \( d_{50} \) term.

The resultant values of \( FL \) (and consequently, \( V_L \)) are significantly below those which would be yielded from Figure 4-5-1.

5.3 EFFECT OF PIPE DIAMETER ON LIMITING VELOCITY

As shown in Durand’s equation, the limiting velocity generally increases with the square root of the pipe diameter for any given concentration and particle size.
NET POSITIVE SUCTION HEAD (NPSH)

6.1 GENERAL NOTES

One factor limiting the suction performance of a centrifugal Pump is the Net Positive Suction Head (NPSH), required at the pump intake, to avoid cavitation.

a. NPSH Required

The NPSH required by a centrifugal Pump, at any given point on the Head/Quantity (H/Q) curve, is the minimum net amount of energy, (expressed in metres head above absolute zero pressure), that the fluid must have at the entrance to the impeller, to avoid cavitation.

Cavitation is the formation of bubbles of vapour at points where the net positive head falls below the vapour pressure of the liquid. The subsequent collapse of these bubbles, as they flow with the liquid into a zone of higher head, may cause severe erosion of the impeller.

The lowest head in a centrifugal pump occurs behind the leading edge of the vanes in the “eye” of the impeller.

Formation of vapour pockets at these points has the following effects on the pump performance:

i. The head developed decreases.

ii. The efficiency drops.

iii. Rumbling or crackling noises and vibration are produced, sometimes resulting in mechanical failures

iv. The impeller can be subjected to excessive erosion.

Cavitation is a term which is often wrongly applied to conditions of malfunction of a pump, for example, when air is induced into the pump through leaking pipework or when air is induced at the intake to the pump.

Classical references to cavitation in water pumps indicate that, with a given suction system, the pump performance follows the normal H/Q curve from shut-off head to where cavitation commences at a certain flow rate. Beyond this flow rate, the H/Q curve (for that suction system) falls off sharply and drops almost vertically to complete failure of pump performance.

Tests show that this is not the case with slurry pumps, as the H/Q curve does not fall sharply after commencement of cavitation, but falls away gradually from the cavitation-free curve. This is probably due to the use of wide impellers. Vapour bubbles do not form across the whole width of the impeller and the flow is only partly restricted. Total performance failure does not occur as sharply as it does with narrower water pump impellers.

The Net Positive Head (NPH) at a point in a pipeline is the absolute pressure head at that point, plus the velocity head, less the vapour pressure.

Thus, if a pressure head gauge reading is obtained at a point in a pipeline, the NPH at that point is equal to the gauge head reading, plus atmospheric pressure head, minus the liquid vapour pressure head, plus the velocity...