WARMAN INTERNATIONAL LTD

WARMAN
SLURRY PUMPING
HANDBOOK

Australasian Version: Feb 2000

WARNINGS

IMPORTANT SAFETY INFORMATION

•

The **WARMAN PUMP** is both a **PRESSURE VESSEL** and a piece of **ROTATING EQUIPMENT.** All standard safety precautions for such equipment should be followed before and during installation, operation and maintenance.

•

For **AUXILIARY EQUIPMENT** (motors, belt drives, couplings, gear reducers, variable speed drives, etc.) standard safety precautions should be followed and appropriate instruction manuals consulted before and during installation, operation, ***adjustment*** and maintenance.

All guards for rotating parts must be correctly fitted before operating the pump including guards removed temporarily for gland inspection and adjustment.

•

DRIVER ROTATION MUST BE CHECKED before belts or couplings are connected. Personnel injury and damage could result from operating the pump in the wrong direction.

•

DO NOT OPERATE THE PUMP AT LOW OR ZERO FLOW CONDITIONS FOR PROLONGED PERIODS, OR UNDER ANY CIRCUMSTANCES THAT COULD CAUSE THE PUMPING LIQUID TO VAPORISE. Personnel injury and equipment damage could result from the pressure created.

•

DO NOT APPLY HEAT TO IMPELLER BOSS OR NOSE in an effort to loosen the impeller thread prior to impeller removal. Personnel injury and equipment damage could result from the impeller shattering or exploding when the heat is applied.

•

DO NOT FEED VERY HOT OR VERY COLD LIQUID into a pump which is at ambient temperature. Thermal shock may cause the pump casing to crack.

•

FOR THE SAFETY OF OPERATING PERSONNEL, please note that the information supplied in this Manual only applies to the fitting of genuine Warman parts and Warman recommended bearings to Warman pumps.

•

LIFTING PUMP COMPONENTS

- Tapped Holes (for Eye Bolts) and Lugs (for Shackles) on Warman Parts are for lifting Individual Parts Only.
- Lifting devices of adequate capacity must be used in conjunction with these assembly and maintenance instructions wherever they are required to be used.
- Sound, safe workshop practices should be applied during all assembly and maintenance work.
- Personnel should never work under suspended loads.

•

FULLY ISOLATE THE PUMP before any maintenance, inspection or troubleshooting involving work on sections which are potentially pressurised (eg casing, gland, connected pipework) or involving work on the mechanical drive system (eg shaft, bearing assembly, coupling):-

- Power to the electric motor must be isolated and tagged out.
- It must be proven that the intake and discharge openings are totally isolated from all potentially pressurised connections and that they are and can only be exposed to atmospheric pressure.

CONTENTS

Section 1: INTRODUCTION

1.1 PURPOSE OF THIS HANDBOOK

This handbook has been compiled to enable you to better evaluate your slurry pumping requirements, and to provide guidelines for selecting the correct slurry pump for your application.

1.2 DEFINITION OF A SLURRY

A slurry can be a mixture of virtually any liquid combined with some solid particles. The combination of the type, size, shape and quantity of the particles together with the nature of transporting liquid determine the exact characteristics and flow properties of the slurry.

1.3 CHARACTERISTICS OF A SLURRY

Slurries can be broadly divided into the two general groups of non-settling or settling types. Non-settling slurries entail very fine particles which can form stable homogeneous mixtures exhibiting increased apparent viscosity. These slurries usually have low wearing properties but require very careful consideration when selecting the correct pump and drive, because they often do not behave in the manner of a normal liquid. When fine solids are present in the slurry in sufficient quantity to cause this change in behaviour away from a normal liquid, they are referred to as being non-Newtonian.

Settling slurries are formed by coarser particles and tend to form an unstable mixture and therefore particular attention must be given to flow and power calculations. These coarser particles tend to have higher wearing properties and form the majority of slurry applications. This type of slurry is also referred to as being heterogeneous.

1.4 WHAT IS A SLURRY PUMP

There are a large number of differing pump types used in the pumping of slurries. Positive displacement and special effect types such as Venturi eductors are used but by far the most common type of slurry pump is the centrifugal pump. The centrifugal slurry pump utilises the centrifugal force generated by a rotating impeller to impart energy to the slurry in the same manner as clear liquid type centrifugal pumps.

However, this is where the similarities end.

Centrifugal slurry pumps need to consider impeller size and design, its ease of maintenance, the type of shaft seal to be used and the choice of the optimum materials. This is needed to withstand wear caused by the abrasive, erosive and often corrosive attack on the materials. Many other important considerations are also required.

The centrifugal slurry pump must be designed to allow the passage of abrasive particles which can at time be extremely large. The largest Warman slurry pump, for example, can pump particles up to 530mm in spherical size.

Slurry pumps therefore need much wider and heavier impellers to accommodate the passage of large particles. They must also be constructed in special materials to withstand the internal wear caused by the solids.

Refer to APPENDIX 2 – MATERIALS for further details on these special materials. To achieve lower operating speeds, slurry pumps are also generally larger in size than a comparable clear liquid pump in order to minimise wear within the pump. Bearings and shafts also need to be much more robust and rigid. Refer APPENDIX 1 – PUMP TYPES to for further details of the various Warman pump types.

1.5 Components of a Slurry Pump

1.5.1 *Impellers*

The impeller is the main rotating component which normally has vanes to impart the centrifugal force to the liquid. Usually, slurry pump impellers have a plain or a Francis type vane (see Figure 1-1).

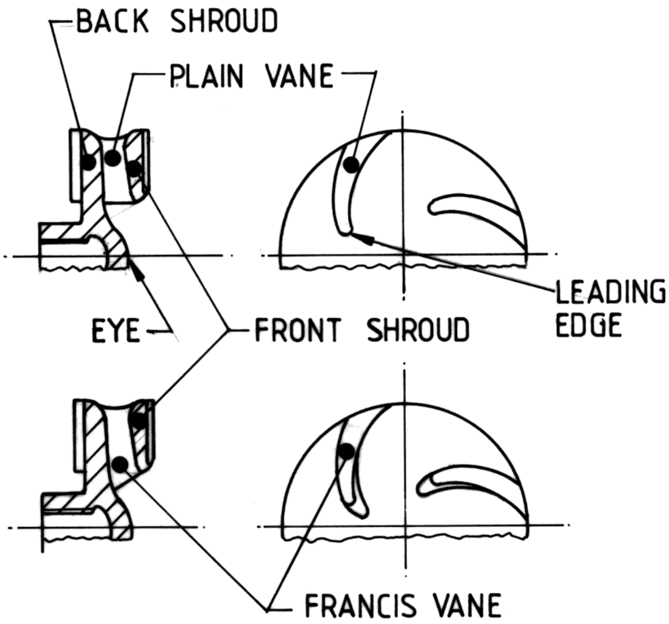


Figure 1-1 Impeller vane profiles

The plain vane has a leading edge square to the back shroud, whereas the Francis vane has a leading edge projecting into the impeller eye.

Some advantage of the Francis vane profile are the higher efficiency, improved suction performance and slightly better wear life in certain types of slurry because the incidence angle to the fluid is more effective.

The plain vane type impeller exhibits better wear life characteristics in very coarse slurry applications or where the mould design precludes the Francis type where an elastomer impeller is required.

The number of impeller vanes usually varies between three and six depending on the size of the particles in the slurry.

Slurry impellers are more commonly of the closed type as illustrated (with a front shroud) but open type impellers (without a front shroud) are sometimes used for special applications.

Impellers are generally closed because of higher efficiencies and are less prone to wear in the front liner region. Open impellers are more common in smaller pumps or where particle blockage may be a problem or where the shear provided by an open impeller is an aid to pumping froth.

Another feature of slurry pump impellers is the pump out or expelling vanes on the back and front shrouds. These perform the dual function of reducing pressure (thus inhibiting recirculating flow black to the impeller eye, and reducing stuffing box pressure) and keeping solids out of the gaps between the casing and impeller by centrifugal action.

The impeller design is crucial as it influences flow patterns and ultimately, wear rates throughout the pump.

The wide range of Warman standard impellers cover most slurry pumping duties or special non-standard designs are also available. Some examples of standard and non-standard impellers are shown in Figure 1-2.

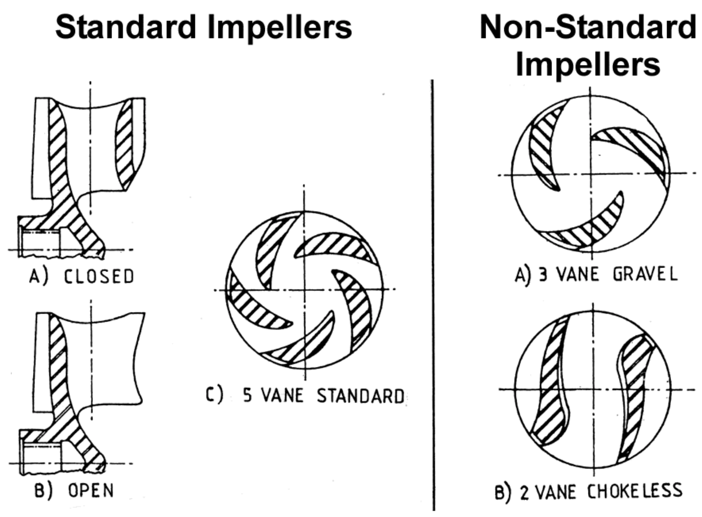


FIGURE 1-2 STANDARD AND NON-STANDARD IMPELLER TYPES

Some typical examples of the need for the non-standard impellers are:

a) Pumping coarse coal

Large particles may cause blockages with a standard 5 vane closed impeller. A special large-particle 4 vane impeller may be required.

b) Pumping fibrous material

Long Fibres may get caught around the vane entrance of standard impellers. A special chokeless impeller can be used for these duties.

c) Reduced diameter impellers

In some special cases, reduced diameter impellers are required but are generally avoided as impeller wear is higher than with full diameter impellers as illustrated in Figure 1-3.

d) Reduced eye impellers

In some extremely high wearing applications such as mill discharge, a special impeller with a reduced eye can prolong impeller wear life.

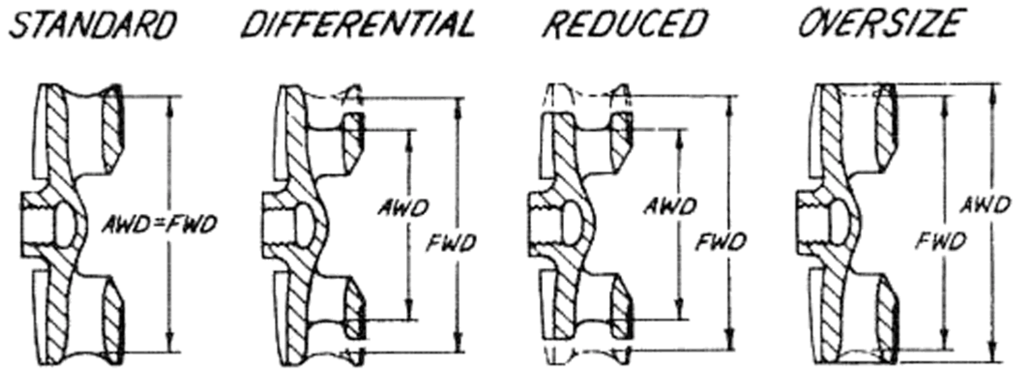


FIGURE 1-3 REDUCED DIAMETER IMPELLERS

1.5.2 CASINGS

Most slurry pump casings are "slower" than their water pump cousins, primarily to reduce wear though lower internal velocities.

The casing shape is generally of a semi-volute or annual geometry, with large clearance at the cutwater. These differences are illustrated in Figure 1-4.

Efficiencies of the more open casings are less than that of the volute type, however, they appear to offer the best compromise in terms of wear life.

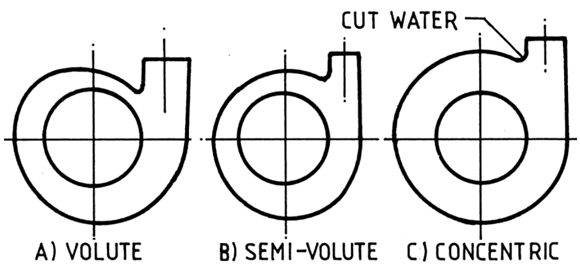


FIGURE 1-4 PUMP CASING SHAPES

1.6 RANGE OF APPLICATIONS OF A SLURRY PUMP

Slurry pumps are used widely throughout the beneficiation section of the mining industry where most plants utilise wet separation systems. These systems usually require the movement of large volumes of slurry throughout the process.

Slurry pumps are also widely used for the disposal of ash from thermal power plants. Other areas where slurry pumps are used include the manufacture of fertilisers, land reclamation, mining by dredges, and the long distance transportation of coal and minerals.

Increased global focus on environmental and energy constraints will certainly generate much wider uses for slurry pumping in years to come.

1.7 CONCEPTS OF MATERIAL SELECTION

Selection of the type of materials to be used for slurry pumping applications is not a precise procedure. The procedure must first account for all the factors (variable characteristics) of the particular slurry. The procedure must take into account the constraints imposed by the following:

a) type of pump,

b) pump speed, and

c) options within the range of the models available.

The basic data required to make a selection of the type of material is:

a) the particle sizing of the solids to be pumped,

b) the shape and hardness of these solids, and

c) the corrosive properties of the "liquid" component of the slurry to be pumped.

The material selection for the pump liners and impellers is made from two basic types of materials:

a) elastomers, and

b) wear/erosion resistant cast alloys

1.7.1 ELASTOMERS

The criteria for selection of the three elastomers commonly used are:

a) Natural Rubber

i) Excellent erosion resistance for liners (against solids up to 15mm size), but limited to particles of 5mm size for impellers.
ii) May not be suitable for very sharp edged solids.
iii) May be damaged by oversized solids or trash.
iv) Impeller peripheral speed should be less than 27.5 m/s, to avoid the thermal breakdown of the liner, adjacent to the outer edge of the impeller. (Special formulations are available to allow speeds up to 32 m/s in certain cases).
v) Unsuitable for oils, solvents or strong acids.
vi) Unsuitable for temperatures in excess of 77°C.

b) Polyurethane

i) Used for pump side liners, where the peripheral speed of the impeller is higher than 27.5 m/s, (and precluding the use of standard rubber) and used for impellers where occasional trash may damage a rubber impeller.

ii) Erosion resistance is greater where erosion is of a sliding bed type rather than one of directional impact. (See Figure 2-2).

iii) Has less erosion resistance to fine solids than natural rubber. Has greater erosion resistance to coarse sharp edged particles than natural rubber, in some circumstances.

iv) Unsuitable for temperatures exceeding 70°C and for concentrated acids and alkalies, ketone, esters, chlorinated and nitro hydrocarbons.

c) Synthetic Elastomers: Neoprene, Butyl, Hypalon, Viton A and others

These are used in special chemical applications under the following conditions:

i) Not as erosion resistant as natural rubber.

ii) Have a greater chemical resistance than natural rubber or polyurethane.*

iii) Generally allows higher operating temperature than natural rubber or polyurethane.*

* Refer to Warman for chemical resistance and temperature limits of individual synthetic rubbers.

1.7.2 WEAR/EROSION RESISTANT CAST ALLOYS

Wear resistant cast alloys are used for slurry pump liners and impellers where conditions are not suited to rubber, such as with coarse or sharp edged particles, or on duties having high impeller peripheral velocities or high operating temperatures.

NOTE: Unlined pumps are generally available only in these types of alloys.

Section 2: DEFINING YOUR APPLICATION & CONSTRAINTS

2.1 PROPERTIES OF A SLURRY

a) Abrasion

Abrasive wear occurs when hard particles are forced against and move relative to a solid surface. Figure 2-1 illustrates the 3 major types of abrasion: Gouging, High Stress Grinding and Low Stress Grinding.

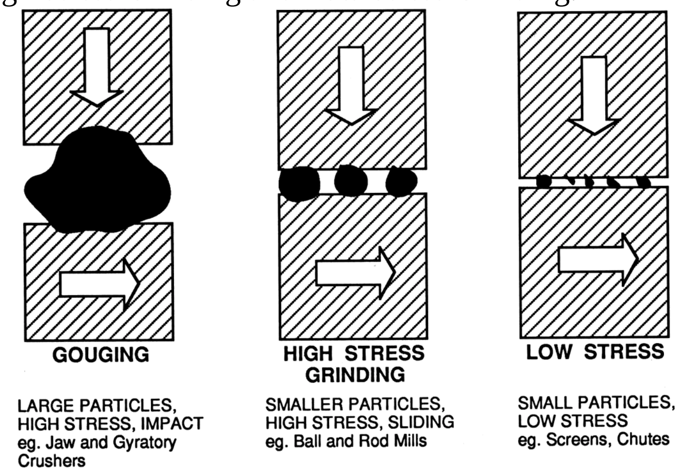


FIGURE 2-1 THREE MAIN MODES OF ABRASIVE WEAR

In a centrifugal slurry pump, abrasion only occurs in two areas:

i) Between the rotating impeller and the stationary throatbush, and

ii) In between the rotating shaft sleeve and the stationary packing.

Abrasion, although used to cover all types of wear, is quite distinct to erosion.

b) Erosion

In slurry pump applications, the dominant mode of wear is erosion. Erosion is a form of wear involving the loss of surface material by the action of particles entrained in the fluid. Erosion involves a transfer of kinetic energy to the particle, which does not occur in abrasion.

The transfer of kinetic energy from the particle to the surface results in a high contact stress. Whilst the overall contact pressure at each impact site is small, the specific contact pressure is high, because of the irregular shape of the particles.

There are three basic types of erosion: sliding bed, random impact and directional impact. These are illustrated in Figure 2-2.

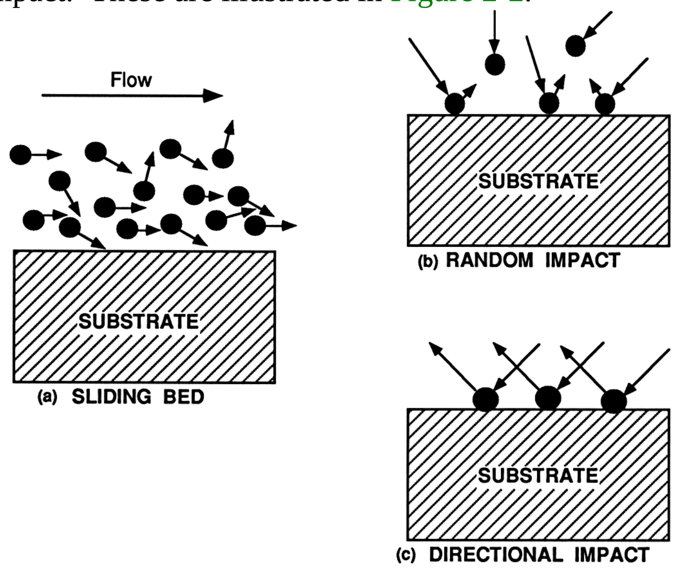


FIGURE 2-2 THREE MAIN MODES OF EROSIVE WEAR

Pump impellers, side liners and volutes wear due to quite different mechanisms as follows:

i) Pump impellers are subjected to a combination of direct impingement (on the leading edge of the vane and at the base of the vane where it joins the back shroud), sliding bed wear and low angle impingement (along the vanes and further inside the passage between the shrouds).

ii) Side liners are mainly subjected to sliding bed wear and some low angle impingement.

iii) Volutes are subjected to direct impingement on the cutwater and sliding bed erosion around the periphery.

c) Corrosion

The corrosion of metals involves the flow of electrical current. Since deterioration depends on electrical factors, as well as chemical factors, it is an extremely complex phenomena. The basis of the corrosion process ($Fe \rightarrow Fe^3$ + 3e-, in the case of iron) corresponds to the action of the anode in an electro-chemical wet battery cell. At the surface of the anode, an electrical charge is transferred from the metal to the liquid (electrolyte). The electrons removed from the region of the anodic reaction flow to the cathode, which then serve as the source of electrons for the cathodic reaction. The electrode reactions vary greatly, depending on the nature of the corroding material and of the corrosive environment.

There are many different types of corrosion, some of which are: uniform; galvanic; crevice; pitting; intergranular; selective leaching; stress and erosion/corrosion. The latter is the most important in slurry applications because the two effects (erosion and corrosion) work together and are often difficult to identify separately.

Elastomers are commonly degraded by many gaseous and liquid environments. Elastomers vary in their ability to absorb a gas or liquid and in their tendency to be dissolved in a solvent. Elastomers dilate when partial absorption or dissolution occurs and this can drastically affect the strength and modulus of elasticity of the material. This results in a deterioration of the wear resistance of the elastomer.

Chemical resistance is a broad term used to describe the deterioration of materials when they are immersed in either a static or slowing fluid.

In the case of an elastomer, chemical resistance may refer to resistance to corrosion or resistance to dilation, and subsequent loss of strength.

d) Solids Concentration

The adverse effects on pump performance caused by solids in a slurry, compared with the pump performance when pumping clear water, are principally due to:

i) Slip between the fluid and the solid particles during acceleration and deceleration of the slurry while entering and leaving the impeller. This slip of solids, and the associated energy loss, increases as the settling velocity of the particles in the slurry increases.

ii) Increased friction losses in the pump. These losses increase with the density (and bulk viscosity) of the slurry.

NOTE: In the following text "Head" (H) is the total head developed by the pump, expressed in metres of the actual liquid or mixture being pumped. For pumping water, we designate the total head developed by the pump as Hw (expressed in metres of water) and for pumping a slurry mixture we use the term Hm (expressed in metres of slurry mixture).

The expression Head Ratio (HR) is the ratio: $\frac{Hm}{Hw}$ where Hm and Hw have the above meanings when the pump handles the same flow rate of water (for Hw) or mixture (for Hm) and the pump speed is the same, in both cases.

The HR is equal to unity for water but decreases as the concentration of solids increases in the slurry mixture. The HR for any given slurry is affected by the particle size and specific gravity of the solids as well as the volumetric concentration of solids in the mixture.

The HR cannot be determined theoretically, but an empirical formula has been developed, from numerous tests and field trials, that allows reliable estimates in most cases.

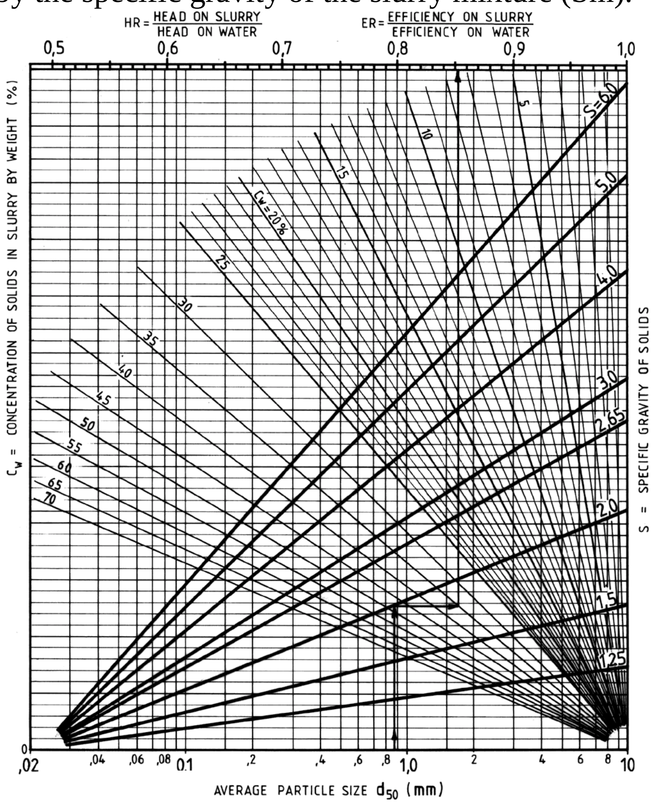
In addition to lowering the head developed by the pump, a rise in solids concentration also reduces the pump efficiency. At high concentrations, this reduction in efficiency could be considerable. For any given pump, it becomes more pronounced with an increase in size of the particles being pumped.

NOTE: In the following text, the symbol "ew" is used to indicate the pump efficiency when pumping water whilst "em" denotes the pump efficiency when pumping a slurry mixture.

The expression Efficiency Ratio (ER) is the ratio: $\frac{em}{ew}$ when the pump is handling the same flow rate of water or of slurry mixture and the pump speed is the same in both cases.

Figure 2-3 has been developed, from test and field results, to enable a reasonable estimation of HR and ER in most practical cases. Using this chart, the pump speed required by a centrifugal pump, when pumping a slurry mixture, will be higher than that indicated by the clear water performance curves.

Similarly, the power required by a centrifugal pump, pumping a slurry mixture will be higher than the value obtained by simply multiplying the clear water power value, by the specific gravity of the slurry mixture (Sm).



This chart applies to mixtures of Solids and Water only.
For mixtures of Solids and Heavy Media or with Liquids
other than Water refer to Warman Office.

FIGURE 2-3 PERFORMANCE OF CENTRIFUGAL PUMPS ON SLURRY

NOTE: This chart applies to simple mixtures of ***SOLIDS AND WATER ONLY***.

e) Effects on Material Selection

The properties of the slurry have a direct relationship to the types of materials required for the components within the slurry pump. Further details on the effects of slurry properties on various types of materials can be found in Concepts of Material Selection. For details of the various material options available, refer to APPENDIX 2 – MATERIALS.

2.2 VOLUME/FLOW RATE

The volume of slurry to be transported must be reliably determined before defining a slurry pumping application. Without a clear understanding of the volumetric requirement and possible variations of demand, it would be impossible to adequately compute a pumping system solution. For slurry pumping, the flow rate is determined by a correlation between three factors:

a) the solids SG,
b) the tonnage of solids required to be pumped, and
c) the concentration of these solids within the slurry mixture.

These three factors need to be determined prior to selecting any slurry pump. An example of how the flow rate can be calculated, using these values, is given in Select Pump Type & Materials.

2.3 PIPELINE LENGTH

Another prime requisite to the evaluation of a slurry pump system is the determination of the length of the pipeline to be used in the application. Slurry passing through a pipeline creates friction (or drag), against the pipe walls. The longer the pipeline, the greater the friction force to be overcome by the slurry pump. Prior to any pump selection, it is therefore imperative that the actual length of the pipeline, and details of any bends or other pipe variations be established, as accurately as possible. Further details on the calculation of pipeline friction can be found in APPENDIX 3 – SLURRY FRICTION HEAD LOSSES IN PIPELINES.

2.4 STATIC HEAD REQUIRED

The actual vertical height (static head) over which the slurry is to be lifted must also be accurately determined prior to selecting a pump. This is relatively easy in plant situations, where the vertical heights involved can be measured or obtained from drawings. In the case of overland pipelines, surveying data is often required to obtain this vital information. Variations in the vertical height (normally measured from the liquid level on the intake side of the pump to the discharge point) can have a major impact on the output of any centrifugal pump. It is therefore important that this vertical height (static head) be determined within reasonable accuracy (0.5m) prior to pump selection. Further details in this important element of slurry pumping can be found in APPENDIX 4 – TOTAL DYNAMIC HEAD.

2.5 PIPE SIZE

The selection of the optimum pipe diameter is also of critical importance in any slurry pumping system. The use of a pipe that is too small can result in either insufficient flow rate or excessively high power requirements. By way of example, a typical slurry flow rate of say, 100 litres per second pumped over 1000 metres would generate friction of 1253 metres in a 100mm ID pipe versus only 60 metres in a 150mm ID pipe. Theoretical power consumption would be around 2000 kilowatts for the 100mm pipe compared to only 250 kilowatts for the 150mm pipe.

The velocity at which the slurry is pumped within the pipeline (determined by the flow rate and the pipe diameter) must also be evaluated to ensure sufficient velocity will be available to maintain the solids in suspension, while they are being pumped. If insufficient velocity is available the solid particles will progressively settle within the pipe, ultimately causing a total blockage of the pipe.

For further details on pipe size and selection, refer to APPENDIX 3 – SLURRY FRICTION HEAD LOSSES IN PIPELINES and APPENDIX 5 – LIMITING SETTLING VELOCITY.

2.6 PUMP PERFORMANCE GRAPHS

To understand the performance of a centrifugal pump, it is necessary to understand how the performance of individual pumps are determined and presented.

Centrifugal slurry performance is usually presented in the form of a performance graph with the flow rate and the head being plotted for a constant speed. Every individual pump model is subjected to a performance test (normally using clear water) at various speeds to enable the composition of a performance graph showing its full range of capabilities.

A typical pump test performance graph is shown in Figure 2-4.

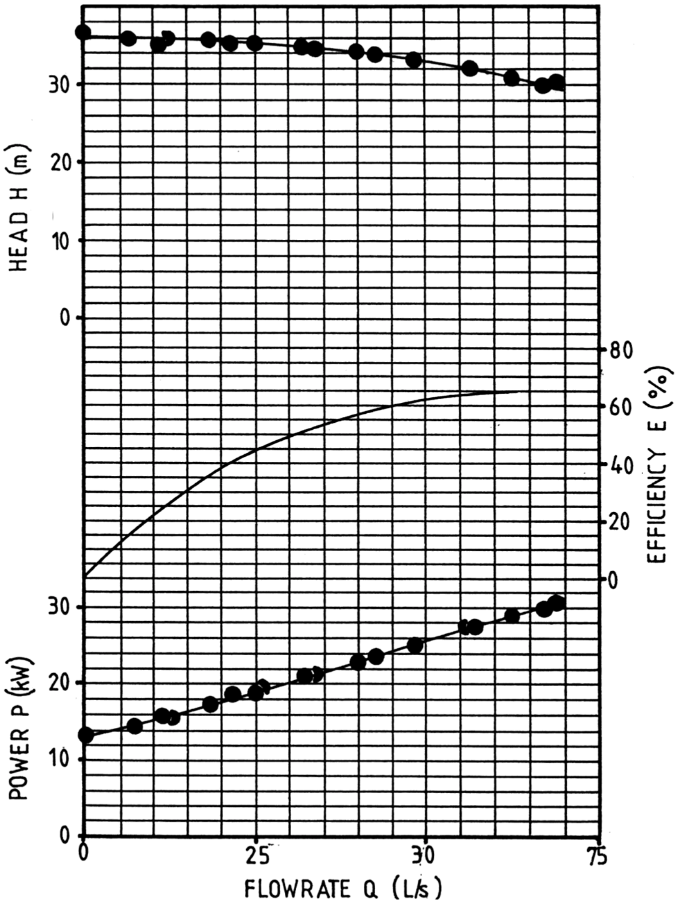


FIGURE 2-4 TYPICAL PUMP PERFORMANCE TEST GRAPH ON WATER

A typical pump performance curve, as issued by Warman, is shown in Figure 2-5.

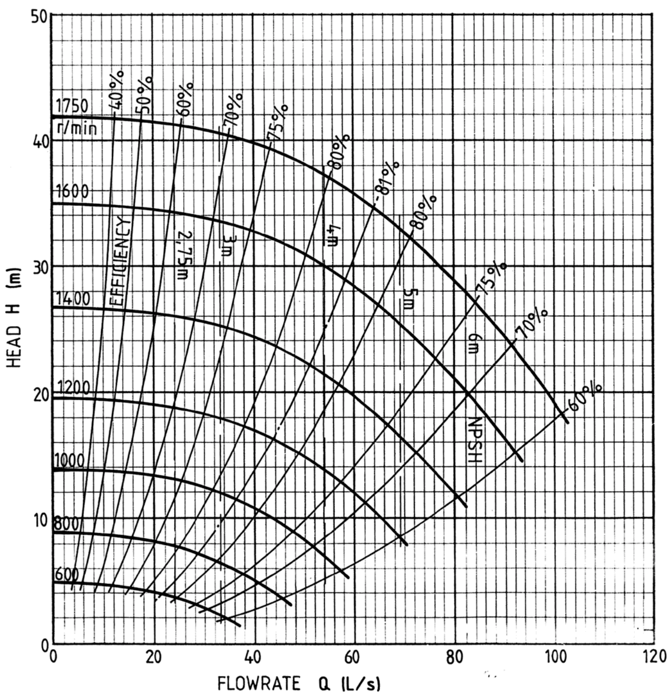


FIGURE 2-5 TYPICAL PUMP PERFORMANCE GRAPH

2.7 SYSTEM RESISTANCE CURVES

The characteristics of a centrifugal pump do not allow a fixed capacity output (as with positive displacement pumps) but rather balance the output against the pipe system. The friction in any pipe system increases with flow rate and can be plotted on what is known as a system resistance curve, as shown in Figure 2-6.

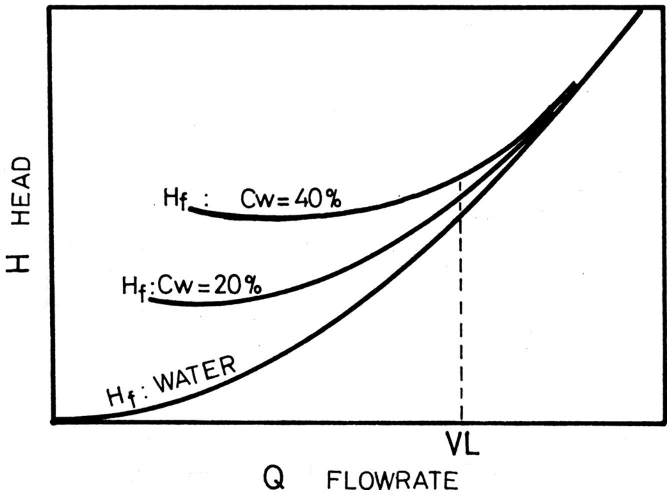


FIGURE 2-6 TYPICAL SYSTEM RESISTANCE CURVE

The intersection of the pump performance curve and the pipe system resistance curve determines the actual pump duty point at which the pump will operate.

This is demonstrated in Figure 2-7.

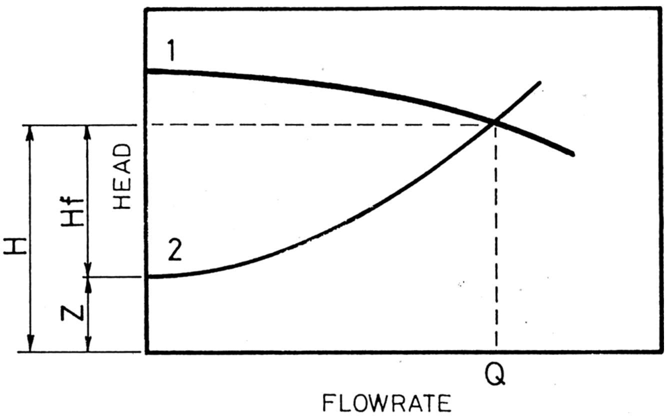


FIGURE 2-7 TYPICAL DUTY POINT CURVE

The pipe system can be defined as all the piping, fittings and devices between the free surface liquid level on the intake side of the pump, to the point of free discharge at the output end of the pipe.

Details on determining the relevant losses that occur in any given system are shown in APPENDIX 3 – SLURRY FRICTION HEAD LOSSES IN PIPELINES.

Centrifugal slurry pumps must overcome both the static head and the system resistance to achieve the movement of slurry to the output end of the pipe system.

The friction losses that occur in any given system can be computed against increasing flow rates and plotted against flow rate and head to generate the system resistance curve.

This system resistance curve is in fact peculiar to any particular piping system, and cannot change unless something in the pipe system is changed, for example:

a) increasing or decreasing the length of pipeline,

b) varying the diameter of the pipe, or

c) varying the static had.

Friction loss is usually established for water, and a correction is made to account for variations in the slurry concentration as described in Appendix 3.

It is important that the system resistance curve be determined when evaluating any slurry pump application, to enable the duty point and potential flow rate variations to be assessed correctly.

System graphs, such as Figure 2-8 and Figure 2-9 are helpful in determining the effects of altering the pump speed or altering some aspect of the pipe system.

Figure 2-8 demonstrates the change in flow rate caused by changing the pump speed.

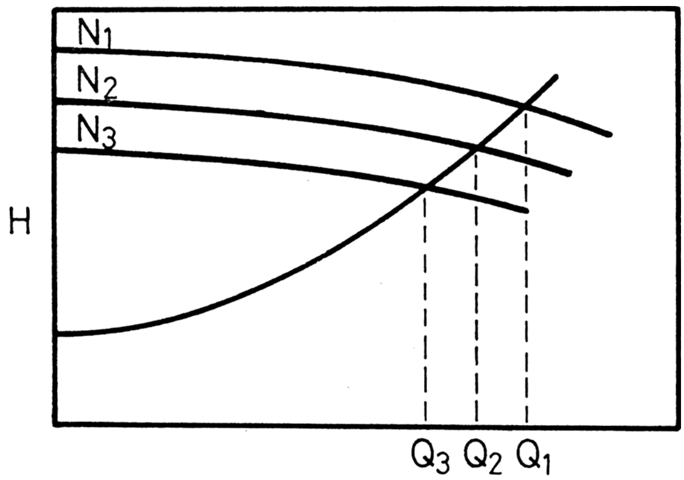


FIGURE 2-8 TYPICAL GRAPH SHOWING PUMP SPEED VARIATIONS

Figure 2-9 demonstrates the change in flow rate caused by changing some aspect of the pipe system.

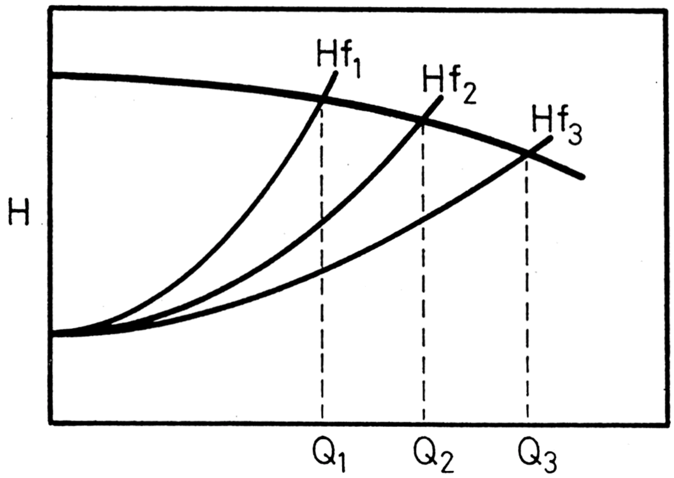


FIGURE 2-9 TYPICAL GRAPH SHOWING SYSTEM VARIATIONS

2.8 OTHER DESIGN CONSTRAINTS

a) Shaft Sealing

The shaft seal is one of the most important mechanical elements in any centrifugal slurry pump and the correct type of seal must be carefully selected to suit each individual pump system. The three most commonly used seal types are as follows:

i) Centrifugal (or Dynamic) Seal

The centrifugal seal is a dynamic, dry seal that only operates whilst the pump is rotating and has no seal effect when the pump is stationary. A secondary seal maintains the liquid within the pump when it is stationary. The secondary seal can either be rubber lip seals of grease lubricated packing as illustrated in Figure 2-10.

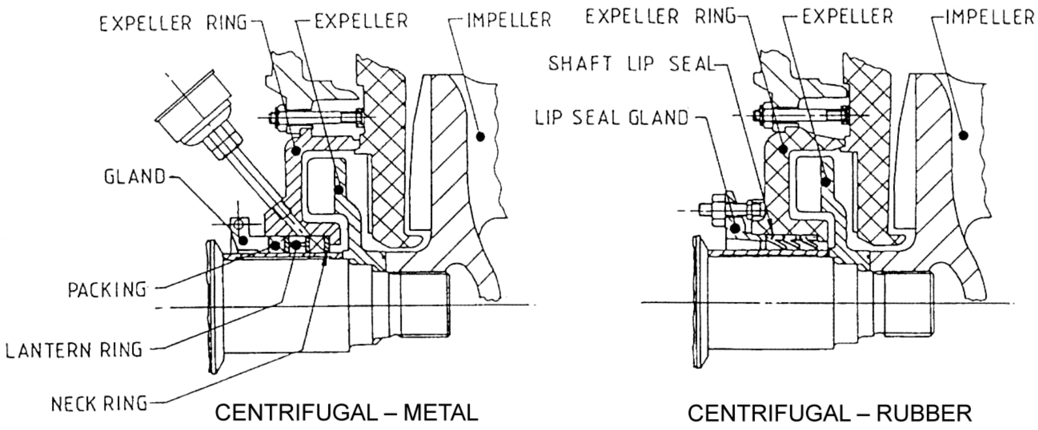


FIGURE 2-10 CENTRIFUGAL (OR DYNAMIC) SEAL ARRANGEMENTS

The centrifugal seal consists of expelling vanes on the back of the impeller and an expeller which rotates in unison with the impeller located in a separate chamber behind the impeller. The expeller acts as a turbine to reduce the pressure of the slurry attempting to escape around the back of the impeller. The expeller forms a pressure ring within the expeller chamber and prevents the slurry from passing into the secondary seal area.

The centrifugal seal is the most common seal used in slurry applications, due to its effectiveness and simplicity, but it is limited by the pump inlet pressure and the pump speed (rpm). Performance data is available for centrifugal seal limitations for specific pump sizes generally as show in Figure 2-11.

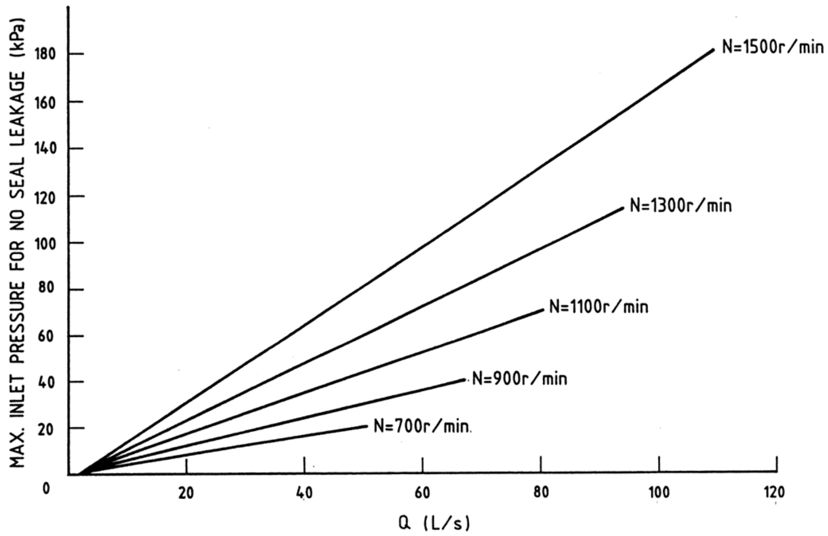


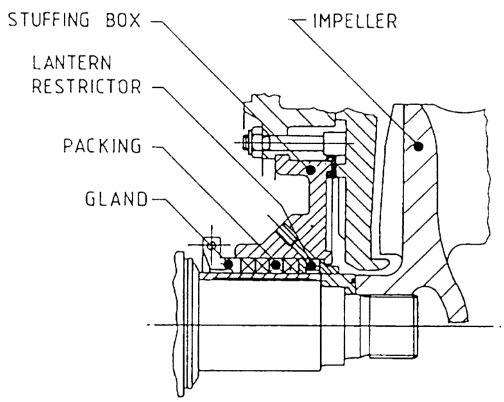
FIGURE 2-11 TYPICAL CENTRIFUGAL SEAL PERFORMANCE CURVE

ii) Gland

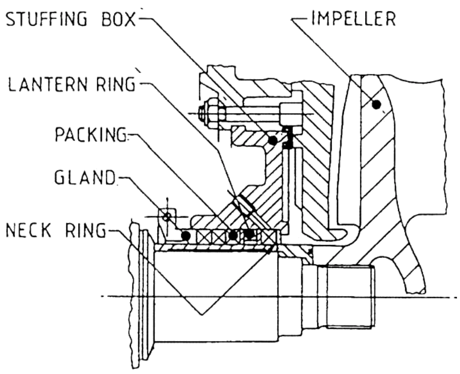
The soft packed gland seal is the second most commonly used seal in slurry applications. The gland seal comprises a number of soft packing rings, compressed in a chamber (stuffing box) against a protective wear sleeve which is fitted to the pump shaft. This type of seal requires continuous liquid lubrication and cooling between the rotating shaft sleeve and the compressed packing, to prevent over heating due to the friction.

The slurry is not a suitable liquid to provide this function, as the particles would very quickly wear through the protective shaft sleeve. A supplementary external supply of clean flushing water must be provided, to flush the slurry particles away from the seal area, whilst providing the necessary lubrication and cooling required by the packing. The quality, quantity and pressure of this gland sealing water is of prime importance and must be carefully matched to the duty required.

Two alternate gland arrangements are shown in Figure 2-12.



WATER SEAL – FULL FLOW



WATER SEAL – RESTRICTED FLOW

FIGURE 2-12 GLAND SEAL ARRANGEMENTS

iii) Mechanical Seal

Mechanical seals are not widely used in slurry applications, but their use in special circumstances is increasing. The mechanical seal consists of a stationary and a rotating face pressed together under mechanical and hydraulic pressure, to prevent leakage.

Alpha grade silicon carbide is the most common material used for manufacture of these seal faces.

The use of mechanical seals in slurry applications requires extreme care and attention due to the limited reliability common in this developing area. Seal costs are relatively high and require substantial justification to warrant their use.

Development of this type of seal however, is being actively carried out by Warman and other seal specialists and it is expected that greater reliability and lower production costs will lead to an increase in their use. Applications where a centrifugal seal cannot be used, and where the addition of water cannot be tolerated, provide the most likely areas for the use of mechanical seals.

A typical seal arrangement is shown in Figure 2-13.

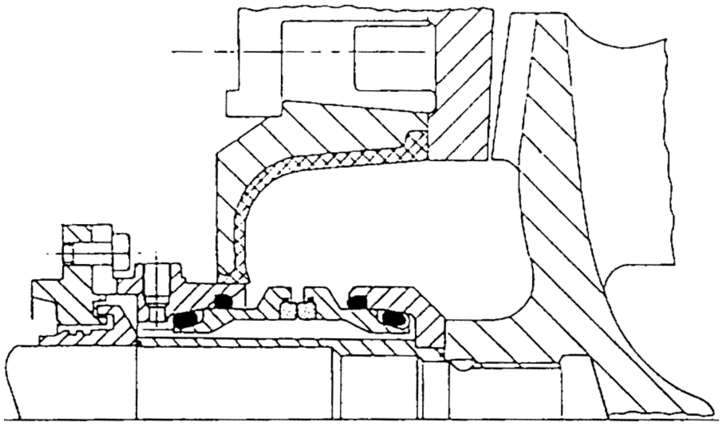


FIGURE 2-13 TYPICAL MECHANICAL SLURRY SEAL

b) Pump Hoppers

It is often the case for low to medium head duties, where the head and quantity requirement is fixed (or nearly so), to operate the pump at a fixed speed and allow the liquid level on the intake side of the pump to vary naturally.

The variation in liquid level is usually made possible by the use of a pump hopper or some other form of feed tank.

Figure 2-14 illustrates a typical hopper feed system and the natural flow control principle.

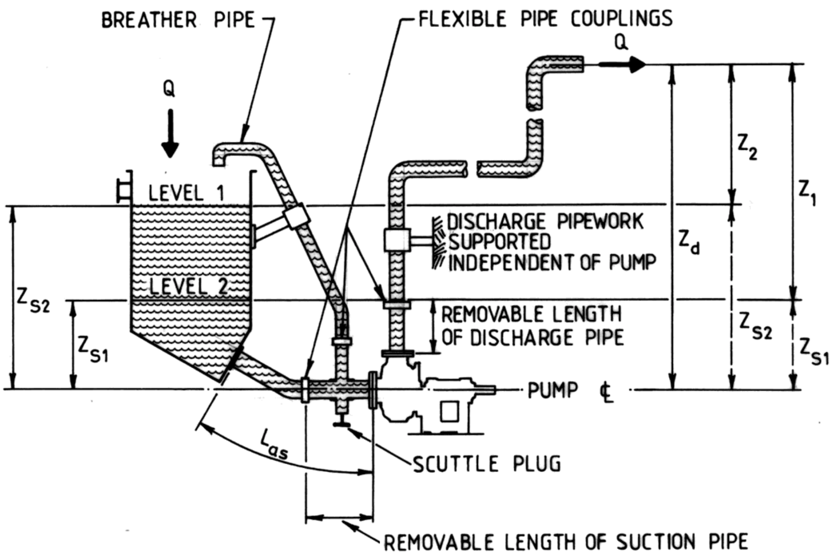


FIGURE 2-14 TYPICAL PUMP HOPPER AND NATURAL FLOW CONTROL PRINCIPLE

Important features of the hopper design are as follows:

i) The height must be sufficient to provide an adequate reserve.

ii) The bottom must be sloped at a minimum of 30°, to hinder the accumulation of settled solids.

iii) The liquid surface area must be sufficiently large to allow a continuous release of entrained air or froth at the free liquid level.

iv) The outlet axis at the base of the hopper should be sloped at a minimum of 30°, to allow air in the suction pipe to be easily displaced (particularly on startup).

v) The suction pipe should be as short as possible to facilitate the displacement of air on startup after the pump has been off-line or after the pump has lost its prime.

vi) The suction pipe should also incorporate a removable, flexible coupling of sufficient distance from the pump flange, to provide access to the pump for maintenance. The support for the remainder of the pipework should be independent of the pump.

vii) A breather pipe is recommended and other special considerations should be made, when the pump is to handle aerated, frothy or very viscous slurries (refer **Air Locks** paragraph for more details).

viii) The suction pipe should incorporate a scuttle plug branch in order to drain the pump and the hopper. A full range of standard Warman hoppers are available to suit each size and type of Warman pump.

c) Air Locks

Horizontal pumps which are gravity-fed from a conventional hopper filled with frothed slurry, will operate in an unstable (cyclic) manner. The output of the pump will oscillate between full and zero flow rate.

This cyclic performance is caused by intermittent air locking. The centrifugal action of the impeller selectively centrifuges slurry away from the eye of the impeller, leaving a growing air bubble trapped at the eye. This accumulation of air impedes the movement of froth from the hopper into the pump, and eventually the pump flow rate will reduce to zero. Consequently, the intake liquid level increases until it is sufficient to compress this air bubble, allowing the froth to reach the impeller, and full flow rate is restored. Air will again begin to accumulate repeating the cycle.

If the intake liquid level in the feed hopper is insufficient to compress the entrapped air bubble, then flow through the pump will not restart until the pump is stopped long enough to allow the bubble to escape.

This tendency to produce air locks may be avoided, or minimised, by providing a vent pipe to allow the trapped air to be released continuously, as shown in Figure 2-15.

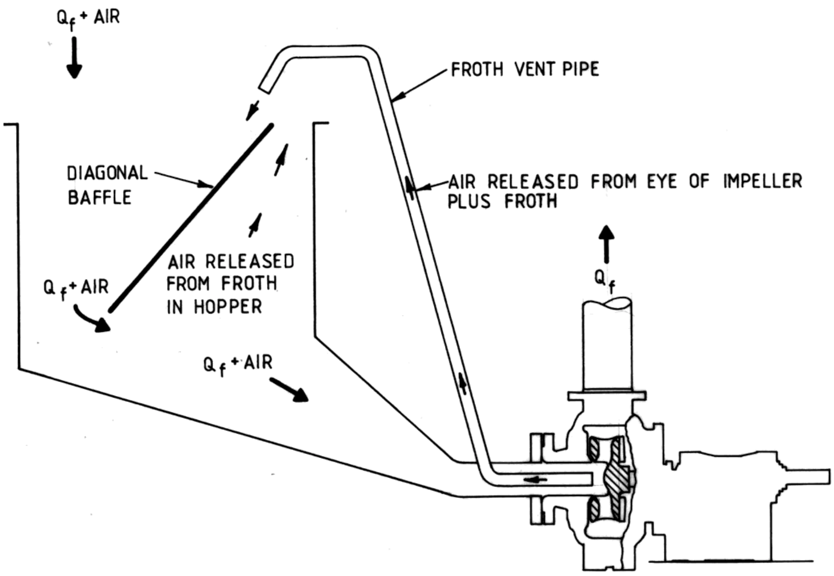


FIGURE 2-15 TYPICAL PUMP HOPPER ARRANGEMENT FOR AERATE OR FROTHY SLURRIES

This arrangement is similar to a normal vent pipe installed, except that the froth vent pipe is extended into the eye of the impeller (to reach the air bubble held by the centrifugal action). The hopper is generally oversized to increase the pressure on the entrapped air bubble.

Sometimes a diagonal baffle is also fitted to the hopper to minimise the regeneration of froth and to assist the escaping air. The feed pipe from the hopper should be extended to a large conical or pyramid shape, to provide an increased entry area for the froth, as close as possible to the pump.

Another solution may be to index the pump heads to the +315° + 270° disposition, see Figure 2-15, which prevents the entrapment of air (in the upper portion of the casing) by the cutwater. This trapped pocket of air would be displaced towards the eye of the impeller when the pump is started if "Standard Vertical" or other dispositions are selected, see Figure 2-16.

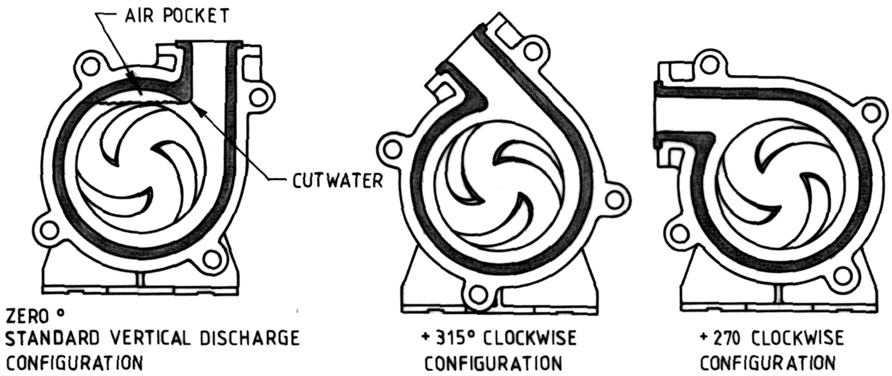


FIGURE 2-16 PUMP DISCHARGE ORIENTATION TO MINIMISE AIR LOCKING

d) Head Loss at Exit into Pressure-Fed Equipment

The exit velocity head, Hve, must be treated as a Head Loss, when the slurry is discharged under pressure into Pressure-Fed equipment, such as hydraulic cyclones (see Figure 2-17) or filter-presses.

Hd = *Zd + Hfd + Hve + Hpf*
= *Hgd + Hvd*
where Hve = *The velocity head in the pipe at the measurement point of the Gauge Pressure Head, Hpf.*

NOTE: If the value of Hpf is specified the value must allow for the head losses downstream of the point of evaluation of Hpf.

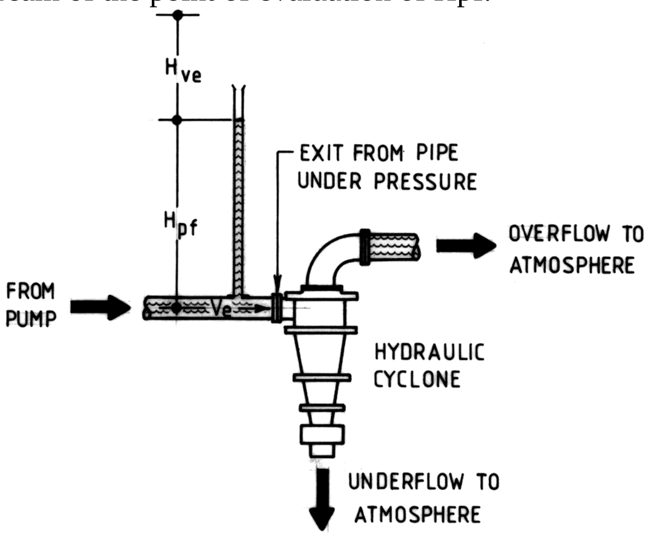


FIGURE 2-17 TYPICAL CYCLONE ARRANGEMENT

e) Pump Burst Hazard

The potential hazard presented by operating any centrifugal pump, whilst the intake pipe and the discharge pipe are simultaneously blocked, is generally well known. The resultant heat generated can result in vaporisation of the entrapped liquid, which in extreme cases, has been known to cause violent bursting of the pump casing.

This potential hazard may be increased when centrifugal pumps are used in slurry applications, due to the nature of the material being pumped. The danger is that a slurry mixture is more likely to cause an accumulation of solids which block the pump discharge pipe and may remain undetected. This situation has been known to lead to a blockage in the intake side of the pump. The continued operation of the pump under these circumstances is extremely dangerous. Should your installation be prone to this occurrence, preventive measures should be adopted to forewarn the operators of this situation.

Section 3: SELECTING THE APPROPRIATE PUMP

Prior to selecting the pump, carry out steps 3.1 to 3.7, then follow steps 3.8 to 3.11.

3.1 DETERMINE THE FLOW RATE

The flow rate can be evaluated in numerous ways, but is usually established by the volume of solids to be pumped and the proposed concentration of solids and liquid. An example of calculating the flow rate is given in Typical Pump Calculation.

3.2 DETERMINE THE STATIC HEAD

The static head (vertical height on both the intake and discharge side of the pump) must be established, and the difference calculated to determine the net static head to be overcome by the pump.

3.3 DETERMINE THE PUMP HEAD & EFFICIENCY CORRECTIONS

It is also necessary to determine the effect of the slurry on the performance of the pump. It will be necessary to know:

a) the average particle size, d50, of the solids to be pumped (d50 is the theoretical screen size where 50% would pass and 50% would be retained),

b) the concentration of solids in the liquid (% by weight), and

c) the dry SG of the solids.

These three values can now be entered into the nomograph shown in Figure 2-3, to determine the Head and Efficiency correcting ratio (HR and ER).

3.4 DETERMINE THE PIPE DIAMETER

It is necessary to determine the pipe diameter that will be required to provide the optimum velocity to minimise friction, whilst maintaining the solids in suspension (to prevent the solids from settling out of the flow).

Details can be found in APPENDIX 3 – SLURRY FRICTION HEAD LOSSES IN PIPELINES and APPENDIX 5 – LIMITING SETTLING VELOCITY.

3.5 CALCULATE THE FRICTION HEAD LOSS

The friction loss created by all the various elements of the pump system must now be calculated. Further details on the calculation of the friction loss can be found in APPENDIX 3 – SLURRY FRICTION HEAD LOSSES IN PIPELINES.

3.6 CALCULATE THE TOTAL DYNAMIC HEAD

The total dynamic head can now be calculated. Further details on these calculations are shown in APPENDIX 4 – TOTAL DYNAMIC HEAD.

3.7 SELECT PUMP TYPE & MATERIALS

Prior to the selection of a specific pump size, it is necessary to determine the pump type required and to establish the type of materials needed. A general description of the various types of Warman pumps available can be found in APPENDIX 1 – PUMP TYPES.

The basic concepts used in the selection of various materials can be found in Concepts of Material Selection with details on available materials given in APPENDIX 2 – MATERIALS.

3.8 PUMP SELECTION

A preliminary selection can now be made from the general selection chart for the various pump types supplied by Warman. A typical example of the Warman selection chart for lined, horizontal slurry pumps is shown in Figure 3-1.

Once the preliminary selection is made, the individual performance curve can then be considered.

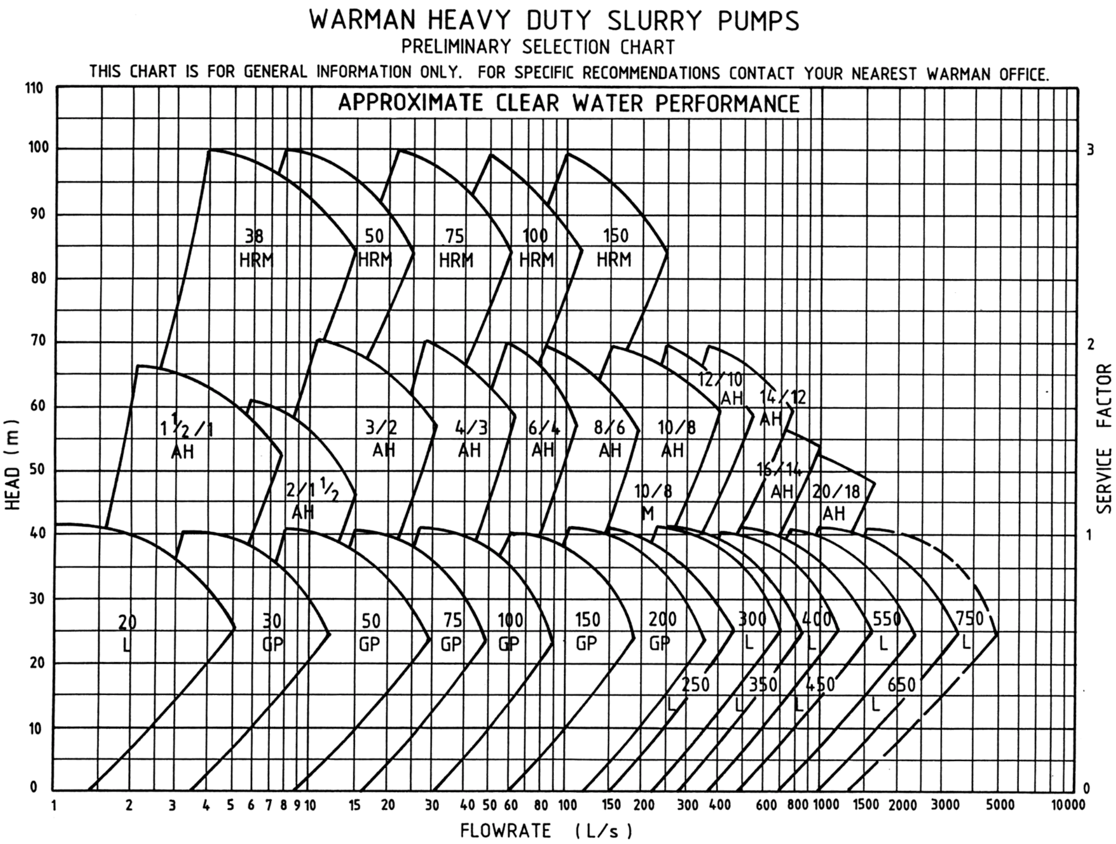


FIGURE 3-1 TYPICAL WARMAN PRELIMINARY SELECTION CHART

NOTE: The service factor indicated is a dimensionless indication of the relative size and wear resistance of the three models depicted.

3.9 DETERMINE THE PUMP SPEED

The speed required can now be determined from the relevant performance curve.

3.10 CALCULATE THE REQUIRED POWER

The power required can now be calculated as shown in the example in Typical Pump Calculation. This will also enable an appropriately sized motor to be selected.

3.11 ADDITIONAL DESIGN CONSIDERATIONS

a) NPSH

The Net Positive Suction Head should be evaluated to ensure that the pump selected will be capable of performing the duty without cavitating. Further data on NPSH can be found in APPENDIX 6 – NET POSITIVE SUCTION HEAD (NPSH).

b) Casting Pressure

It is necessary to calculate the maximum pressure in the pump (usually computed at the pump discharge), to ensure that the maximum pressure limits for the pump casing are not exceeded. Refer to Warman for details of the pressure limits for particular pumps.

c) Froth Pumping

The actual flow rate of froth (slurry PLUS bubbles) to be handled can vary markedly, compared to the flow rate of slurry only (that is, without any bubbles). The Froth Factor may be as low as 120% for a less stable froth, or as high as 150% for a very tenacious froth.

The slurry flow rate, Q, must be multiplied by the Froth Factor to determine the actual froth flow rate, Qf, which is to be handled by the pump. Q and Qf may also vary widely due to the variations in the grade and nature of minerals which are treated, from time to time, in a given flotation plant.

The presence of air bubbles in the froth reduces the effective value of Sm as the froth enters the pump. The value of Sm is quickly increased due to compression of the bubbles by the head developed in the pump. This value is subsequently decreased as the froth passes from the pump, along the discharge pipeline, to the discharge point, which is at atmospheric pressure. Conversely, the value of Vd will increase along the discharge pipeline due to expansion of the bubbles. Further details on considerations required for froth pumping are given in **Air Locks**.

d) Conical Enlargements

In many cases the internal diameter of the discharge pipe may be greater than that at the discharge flange of the pump selected. A conical enlargement section is required to join the pump to the pipework.

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, $\theta = 6°$. In practical terms, this is equivalent to allowing an increase in diameter of 10mm for every 100mm in length of the conical section.

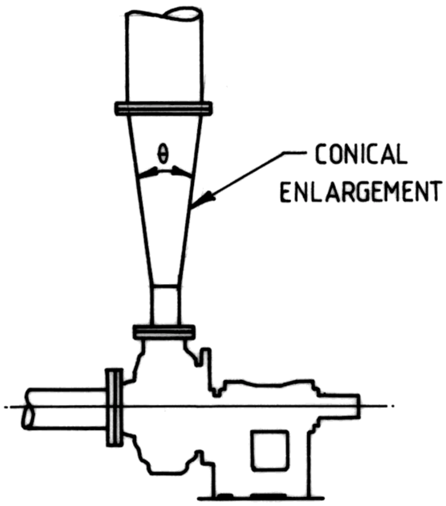


FIGURE 3-2 TYPICAL CONICAL ENLARGEMENT

e) Pump Feed Hoppers

Consideration should be given to the design of a suitable pump feed tank or hopper. Some basic guidelines are given in **Pump Hoppers**.

f) 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 **Shaft Sealing**.

g) Multi-Staging

Where the duty required exceed 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 – SERIES PUMPING.

h) 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 hours of sand

Specific gravity of solids S	=	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.

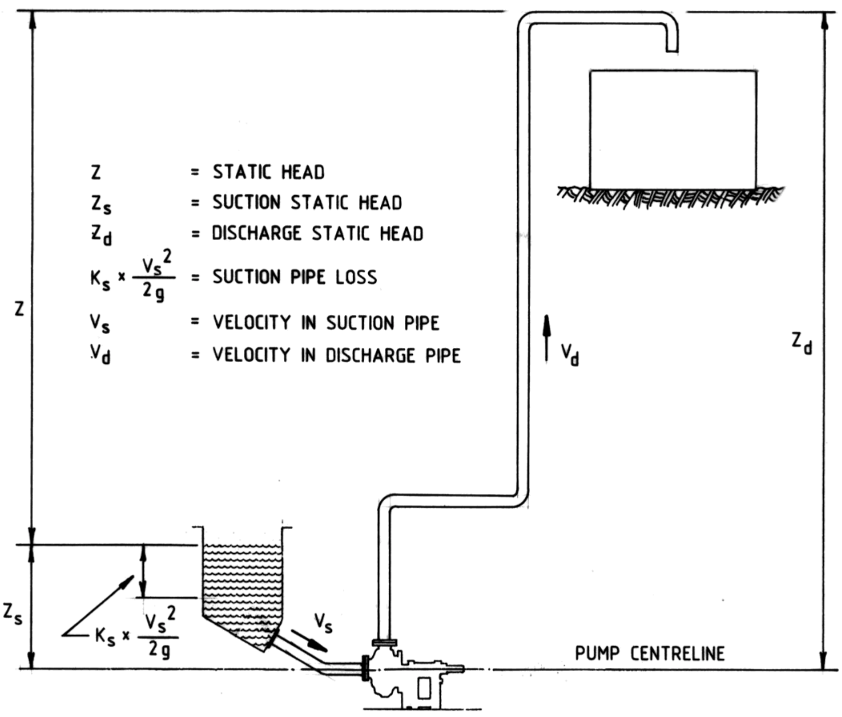


FIGURE 3-3 TYPICAL PUMP APPLICATION

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:

$$Weight\ of\ solids\ in\ slurry = 65\ tonnes$$

$$Weight\ of\ volume\ of\ water\ equal\ to\ solids\ volume = \frac{65}{2.65} = 24.5\ tonnes$$

$$Weight\ of\ water\ in\ slurry\ of\ Cw = 30\% = \frac{65\,(100-30)}{30} = 151.7\ tonnes$$

$$Total\ weight\ of\ equal\ volume\ of\ water = 151.7 + 24.5 = 176.2\ tonnes$$

$$*\,(1m^3\ of\ H_2 0 = 1\,tonne)$$

$$Total\ weight\ of\ slurry\ mixture = 65 + 151.7 = 216.7\ tonnes$$

$$Specific\ gravity\ of\ slurry\ mixture\ (Sm) = \frac{216.7}{176.2} = 1.23$$

$$Concentration\ of\ solids\ by\ volume\ (Cv) = \frac{100}{176.2}\ x\ 24.5 = 13.9\%$$

$$Quantity\ of\ slurry = 176.2\ m3/hr\,*$$

$$= 49\ L/s$$

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:

$$V = \frac{Q\ x\ 1273}{d^2}$$

where V = *slurry velocity in m/s*
Q = *slurry flowrate in L/s*
d = *pipe diameter in mm*
g = *9.81 m/s²*

Velocity V in this case is therefore:

$$\frac{49\ x\ 1273}{150^2} = 2.8\ m/s$$

Using Durand's equation from APPENDIX 5 – LIMITING SETTLING VELOCITY.

$$VL = FL\sqrt{2gD\left(\frac{S - Sl}{Sl}\right)}$$

where D = *Pipe diameter in m.*

The value of FL is obtained from Figure A5-2, using a Cv of 13.9% and an average particle size d50 = 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:

$$VL = 1.04\sqrt{2\ x\ 9.81\ x\ 0.15\ x\left(\frac{2.65-1}{1}\right)}$$
$$= 2.3\ m/s$$

The 150mm pipe is therefore considered suitable for this application since the limiting settling velocity (2.3 m/s) is lower then 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 Figure A4-3 and Figure A4-4.

Actual length of line	*= 100m*
5 x 90° long radius bends at 3.35 metres each	*= 16.75m*
Equivalent length of line	*= 11.75m*

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 A3-2.

By substitution in Darcy's equation for friction head as in Figure A3-2:

$$Hf = 0.017\ x\ \frac{116.75}{0.15}\ x\ \frac{2.8^2}{2\ x\ 9.81}$$
$$= 5.29m\ of\ mixture\ for\ 116.75m\ of\ pipe$$

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 A4-4. Head loss in this case using an enlargement included angle of 30° would be.

$$Ke\frac{(V - Vl)^2}{2g} = 0.55\ x\ \frac{(6.24-2.4)^2}{2\ x\ 9.81}$$
$$= 0.41m$$

e) Loss at pipe discharge

Under the 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 $\frac{(V^2)}{2g}$

is $\frac{2.8^2}{2\ x\ 9.81} \quad = \quad 0.4m\ of\ \ mixture.$

f) Loss of head at entrance to suction pipe

This is dealt with in Figure A4-4. 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:

$$0.5\ \ x\ \frac{V^2}{2g} \quad = \quad 0.2m$$

g) Total dynamic head on the pump (refer Figure A4-1 and Figure A4-2)

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

$$Hm \quad = \quad Z \ + \ Hf$$
$$where\ Z\ is\ static\ head;\ \ ie, (Zd - Zs)$$
$$Hm \quad = \quad ((20-1) \ + \ 5.29 \ + \ 0.14 \ + 0.4 \ + 0.2)$$
$$= \quad 25.4\,m\,of\ \ slurry\ mixture.$$

h) Equivalent water total dynamic head

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

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

$$\frac{Hm}{HR} \quad = \quad \frac{25.4}{0.89} \quad = \quad 28.53$$

Say, 28.5 m of water

i) Pump Selection

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

Total head of 28.5m of equivalent water and 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 3-3) thus:

$$= \frac{Q \ \ x \ \ Hm \ \ x \ \ Sm}{1.02 \ \ x \ \ em}$$

$$or \quad \frac{Q \ \ x \ \ Hw \ \ x \ \ Sm}{1.02 \ \ x \ \ ew} \qquad (as\ HR\ is\ assumed\ equal\ to\ ER)$$

$$= \frac{49 \ \ x \ \ 28.5 \ \ x \ \ 1.23}{1.02 \ \ x \ \ 66} \quad = \quad 25.5\ kW$$

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

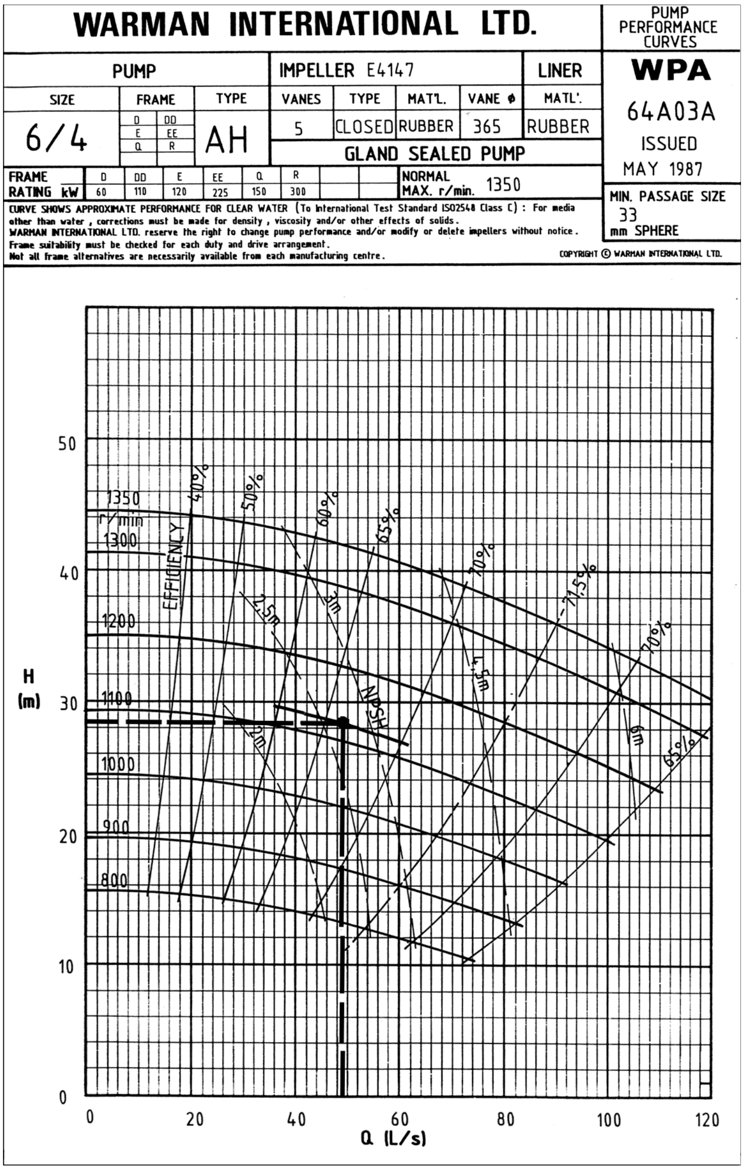


FIGURE 3-4 WARMAN PUMP PERFORMANCE CURVE

APPENDIX 1 – PUMP TYPES

INTRODUCTION

Warman slurry and liquor pumps are generally of the centrifugal type (except for jet pumps) with the range consisting of twenty basic models. General descriptions are as follows.

HORIZONTAL PUMPS - LINED

This group of pumps feature interchangeable casing liners and impellers of elastomer or hard metal material.

a) Type AH and AHE

'AH' and 'AHE' pumps are designed for a wide range of erosive and/or corrosive applications. The 'AHE' pumps utilise enhanced performance components that are retrofittable to the Type 'AH' casings and bearing assemblies.

They are generally used for slurries containing high concentrations of erosive solids or where an extremely robust and heavy duty pump is required.

All Warman standard seal options are available of centrifugal, gland or mechanical types and include the Warman high seal and low flow options.

Pump sizes range from 25mm discharge size through to 450mm.

A range of alternative drive frames is available to allow widely varying power demand requirements.

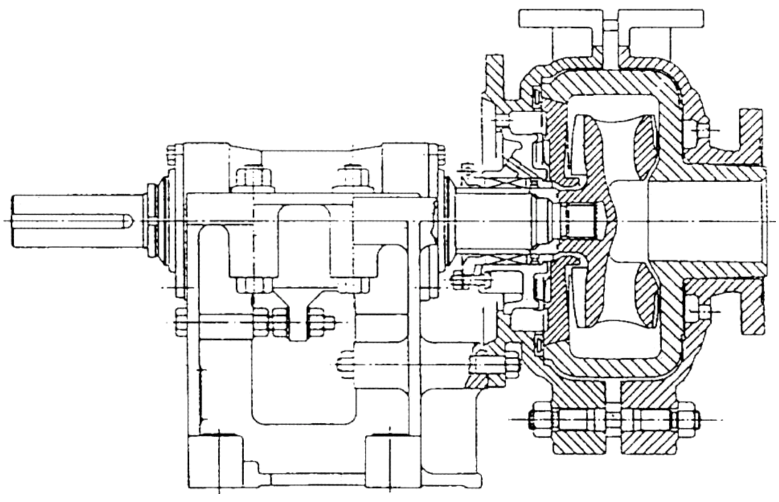


FIGURE A1-1 TYPE AH PUMP

b) Type GP

The Warman 'GP' series is specifically designed to handle a wide range of corrosive and erosive liquors and slurries. They are typically used in chemical applications or where slurries contain lower concentrations (up to 35% by weight) of erosive solids. The 'GP' can also be used for pumping higher concentrations of less erosive solids. Sealing options include centrifugal, gland and mechanical.

The 'GP' is somewhat smaller in size to the corresponding 'AH' model, This range features a high strength fibre reinforced outer casing.

Sizes range from 30mm to 200mm discharge.

The GP features interchangeable bearing frames as with most other Warman models.

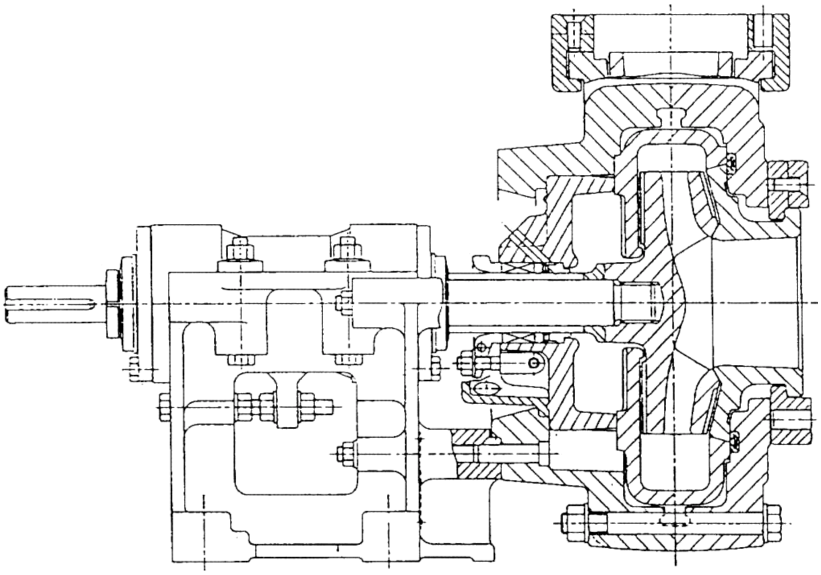


FIGURE A1-2 TYPE GP PUMP

c) Type L

'L' pumps are designed essentially for the same range of applications covered by the 'GP' series, but cover much higher flow ranges with discharge sizes extending up to 650mm.

Alternative drive frame sizes are available in all models to accommodate varying power demand requirements. As with the GP, bearing frames are interchangeable with most other Warman models. Standard Warman seal options are also available.

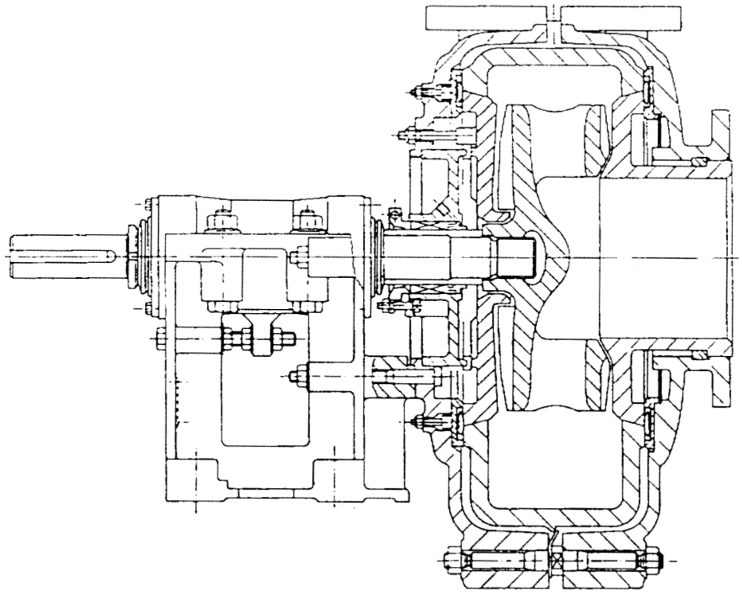


FIGURE A1-3 TYPE L PUMP

d) Type AHP and AHPP

'AHP' and 'AHPP' pumps have the same hydraulic performance as the Type 'AH' pumps and normally use the same liners and impellers. The same performance curves are also applicable. These pumps are used where higher pressure ratings are required, usually in multi-stage pump installations and feature heavily reinforced outer castings to contain high internal pressures (up to 5000kPa).

e) Type HRM

'HRM' pumps are characterised by a larger diameter impeller than the corresponding Type 'AH' pumps. They are used for applications requiring higher heads (up to 95m per stage) or in extremely heavy duty applications requiring very slow impeller speeds.

All standard seal and material options are available although impellers are generally restricted to hard metal options only.

f) Type W

'W' pumps consist of a Type 'AH' pump fitted with a tank between the pump base and the pump wet-end. The tank is filled with water which prevents air entering the pump through the gland when operating with negative suction heads. These pumps are mostly used on vacuum filtrate-extraction duty. The submerged gland seal precludes the use of non-return valves in high vacuum duties and maintains a water tight seal at all times.

Hydraulic performance of these pumps are identical to the corresponding Type 'AH' pumps. For example, the performance for a 3/2 C-AHW pump is the same as the 3/2 C-AH pump.

Available in sizes ranging from 25mm diameter discharge up to 150mm.

g) Type SHD

'SHD' pumps feature advanced hydraulics design for high efficiency and long wear life suitable for the most arduous service, eg. mill discharge service. These pumps feature metal impellers with metal and/or elastomer type liners and/or side liners.

Sizes range from 75mm to 750mm discharge diameter and feature ease of maintenance which is assisted by the use of special purpose assembly tools.

h) Type AHF, MF and LF

These three horizontal types of pumps are for handling froths, in particular tenacious mineral froths. These pumps have large inlets and open impellers to cater for the high percentage of air normally experienced with tenacious froths.

Being horizontal they are smaller than the vertical type AF froth pumps, require less head room and generally are easier to maintain than a vertical pump.

HORIZONTAL PUMPS - UNLINED

This group of pumps feature a metal casing with no separate liners and no elastomer wear parts.

a) Type D

'D' dredge pumps are designed for dredging and similar low head duties. The design features a hard metal casing and wear components and is capable of passing extremely large particles. Sizes range from 350mm diameter discharge up to 920mm.

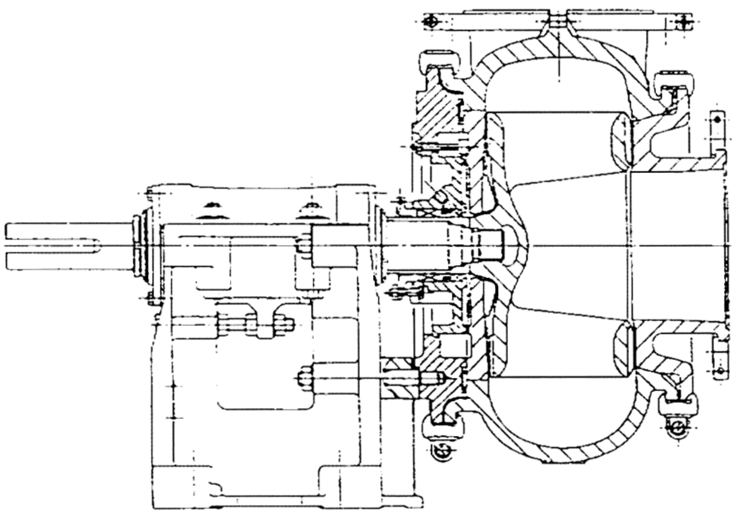


FIGURE A1-4 TYPE D PUMP

b) Type G

Type 'G' gravel pumps are similar in design to the Type 'D' but feature larger impellers and heavier casing construction. They are typically used for pumping gravel, dredging or pumping solids too large to be handled by Type 'AH' pumps.

Sizes range from 100mm up to 600mm.

c) Type GH

The 'GH' range is again similar in construction to both the 'D' and 'G' Types, but features larger impeller diameters than the G range and incorporates a heavily reinforced casing design to allow pumping of heads up to 80 metres. Typically used in dredging applications where long discharge distances are required. Some models are available with higher pressure rating suitable for applications such as multi-staging.

The 'GH' is available in sizes ranging from 150mm diameter discharge up to 400mm.

d) Type S

The 'S' Type solution pump range is an uncased solution pump designed for dirty water and liquor applications. The pump features common drive frame componentry with other Warman models. The 'S' range is available in sizes for 38mm diameter discharge up to 400mm.

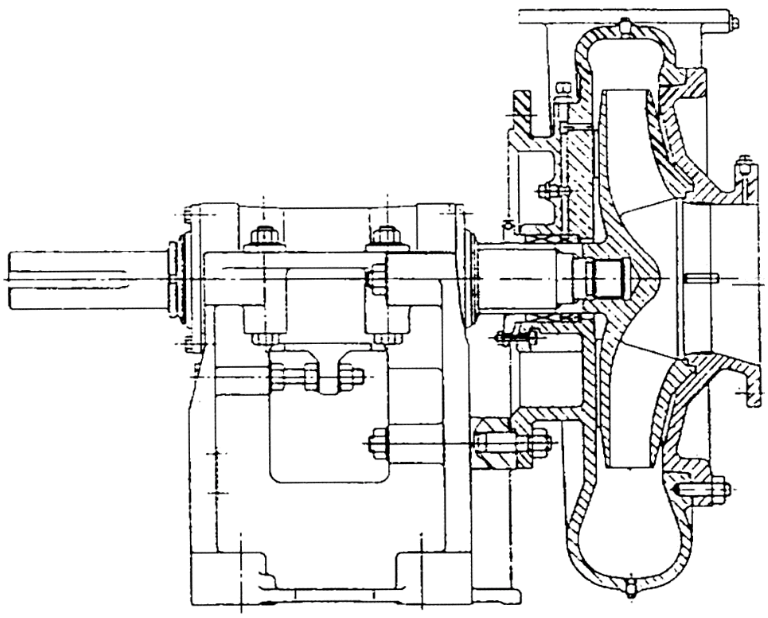


FIGURE A1-5 TYPE S PUMP

e) Type SH

The 'SH' Type solution pump range is similar to the 'S', but features larger impeller diameters and heavily reinforced casing for use where higher heads are required (up to 125m).

The 'SH' range is available in sizes from 50mm diameter discharge up to 250mm.

f) Type PC (and PCH)

The 'PC' range is an uncased hard metal design specifically designed for pumping corrosive and erosive slurries where heavy duty slurry pumps are not warranted but clear liquor pumps are not adequate. The 'PC' range is also available in a high head form ('PCH') suitable for heads up to 120m.

The 'PC' range is available in sizes ranging from 50mm discharge up to 250mm.

Standard seal options featuring high seal expellers or low flow glands are available.

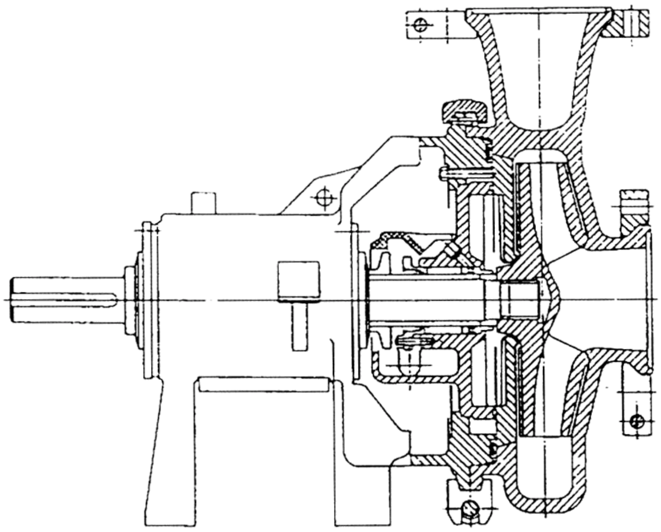


FIGURE A1-6 TYPE PC (PCH) PUMP

g) Type TC

The CYKLO ‘TC’ Type is an uncased design available in erosion resistant materials designed specifically for “non clog” or “gentle” pumping applications such as carbon in pulp processes.

The ‘TC’ range is available in sizes ranging from 50mm diameter discharge up to 200mm diameter discharge.

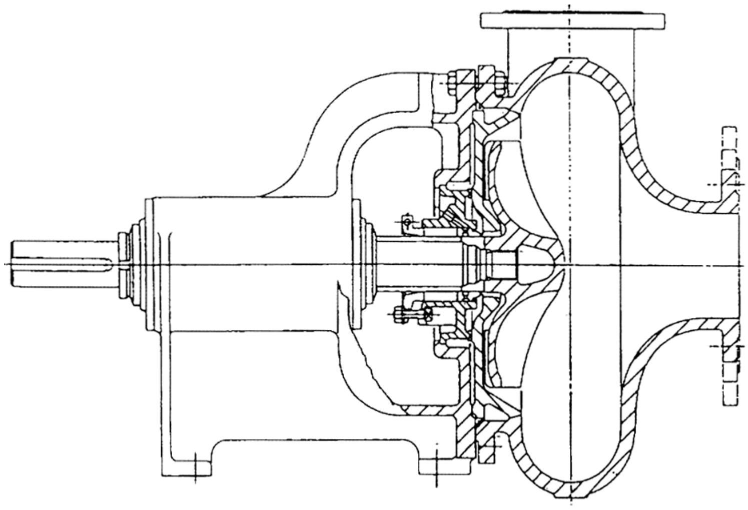


FIGURE A1-7 TYPE TC PUMP

h) Type SHDU

‘SHDU’ pumps are the unlined version of the SHD pump range. These pumps have the same advanced hydraulics and ease of maintenance as the SHD pumps. Some of the wearing parts, eg. impellers, are the same as the SHD components allowing relatively easy conversion from, say, lined to unlined.

VERTICAL PUMPS

a) Type GPS (SP)

The ‘GPS’ range is a vertical cantilevered shaft design available with a variety of erosion resistant wet end materials including hard metal, rubber* and polyurethane*. The pump features a self venting double intake style and does not require submerged bearings. These pumps casing available with self-agitating, non clog impellers. Available in sizes ranging from 40mm through to 250mm.
(* Only available in 40, 65 and 100 sizes).

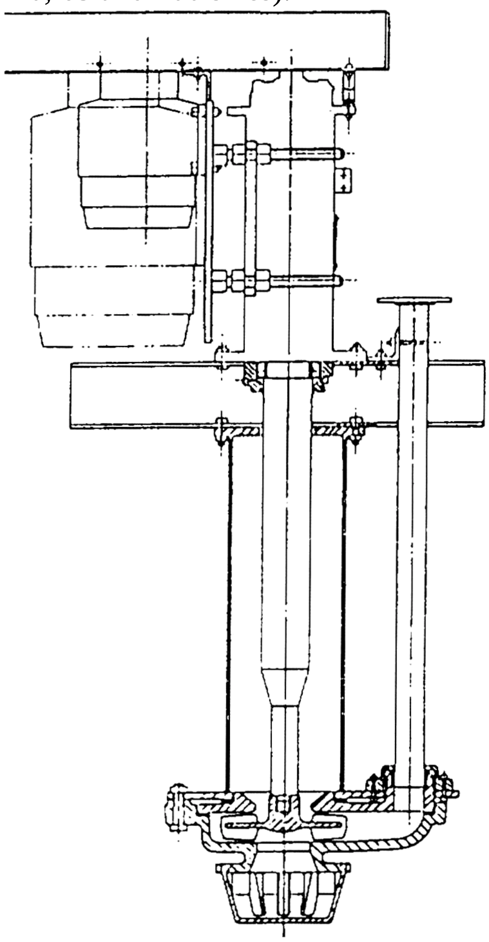


FIGURE A1-8 TYPE GPS PUMP

b) Type SPR

The 'SPR' range is vertical cantilevered shaft design similar to the 'GPS', but features full elastomer protection on all submerged components.

This allows this pump to be used in highly corrosive applications. Available in sizes ranging from 40mm through to 150mm.

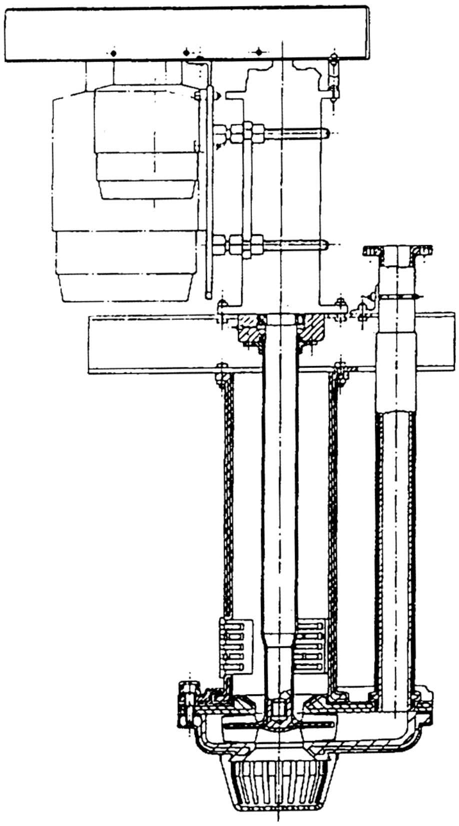


FIGURE A1-9 TYPE SPR PUMP

c) Type V-TC

The 'V-TC' range is a combination of the CYKLO ('TC') Type hard metal wet end fitted to the vertical cantilevered shaft bearing assembly.

These pumps find application where the non clogging or 'gentle' pumping features of the CYKLO design are required in a vertical submerged situation.

Available in discharge diameters ranging from 50mm to 200mm.

d) Type AF

The type ‘AF’ Froth Pump is a vertical pump complete with hopper which utilises casings and liners from the Type ‘AH’ pumps.

It is designed to pump frothy slurries more efficiently by de-aerating or partially de-aerating the froth before it enters the pump head, utilising an induced vortex principal.

The ‘AF’ range is available in sizes ranging from 20mm up to 200mm discharge diameter.

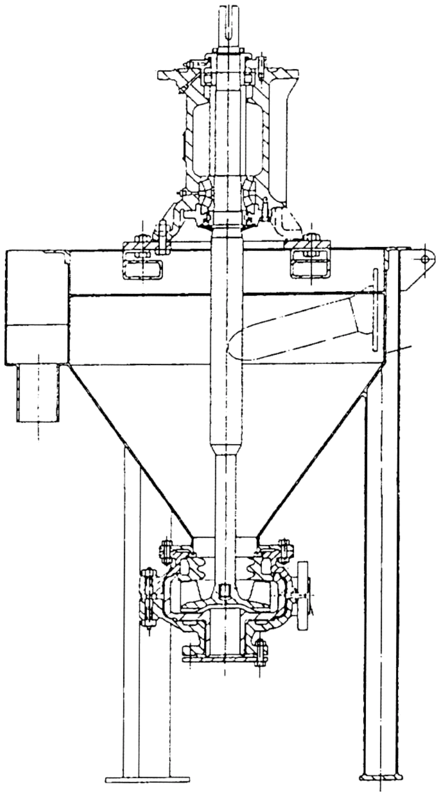


FIGURE A1-10 TYPE AF PUMP

JET PUMPS

Jet pumps, often referred to as eductors or ejectors, differ from other Warman designs in that they have no moving parts. Power is supplied by a high pressure jet of liquid. The Warman jet pumps can be fitted with a variety of erosion resistant materials and can be used in a wide range of special applications. The range covers five different models, ranging from 50mm to 300mm in size.

For further details on individual models generally described, please contact your nearest Warman office.

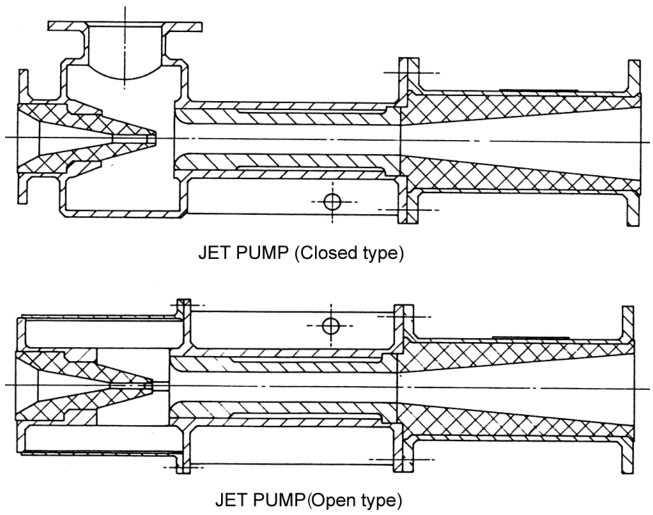


FIGURE A1-11 JET PUMPS

APPENDIX 2 – MATERIALS

INTRODUCTION

A major advantage of the Warman slurry pump is the number of optional materials available. This enables a pump to be constructed with the most appropriate materials specifically to meet the duty requirements. It also allows existing pumps to be adapted in service to meet changed duty conditions, merely by changing individual parts.

A general description of some of the more common materials used in Warman slurry pump construction is listed in Table 1 below.

Further assistance with specific material selections can be obtained from your nearest Warman office.

TABLE 1 MATERIAL SPECIFICATIONS AND DESCRIPTION

WARMAN CODE	MATERIAL NAME	TYPE	DESCRIPTION
A03	Ni-Hard 1	Martensitic White Iron	Alloy A03 is a maternsitic white iron which offers reasonable performance in mildly erosive duties, and where low impact levels are experienced. It is generally heat treated to stress relieve or reduce the amount of residual austenite in the matrix. The alloy is sensitive to section thickness, and the composition requires adjustment to prevent the formation of undesirable phases.
A04	ULTRACHROME 24% Cr	Erosion Resistant White Iron	Alloy A04 is a white iron having a hardness of 375HB in the annealed state. This low hardness allows A04 to be more readily machined than alloy A05. The alloy can be subsequently hardened to increase the wear resistance. A04 is not as erosive resistant as A05 and A12, and is not generally corrosion resistant.
A05	ULTRACHROME 27% Cr	Erosion Resistant White Iron	Alloy A05 is a wear resistant white iron that offers excellent performance under erosive conditions. The alloy can be effectively used in a wide range of slurry types. The high wear resistance of alloy A05 is provided by the presence of hard carbides within its microstructure. Alloy A05 is particularly suited to applications where mild corrosion resistance, as well as erosion resistance is required.
A06	Ni-Hard 4	Marternsitic White Iron	Martensitic wear resistant alloy.

WARMAN CODE	MATERIAL NAME	TYPE	DESCRIPTION
A07	15/3 Chrome/Moly Iron	Chromium/ Molybdenum White Iron	Martensitic white iron with moderate erosion resistance.
A12	HYPERCHROME® 30% Cr	Hypereutectic Chromium White Iron	HYPERCHROME® alloy is a hypereutectic white iron suitable for high wear duties, where corrosion is not considered a problem. It should be used in applications where A05 and A04 do not provide an adequate wear life. Alloy A12 can be used in mild alkaline slurries, between a pH range of 8 to 14. The alloy may provide up to three times the wear life of A05 and A03 parts in some severe applications.
A14	ULTRACHOME Tough 27% Cr	Erosion Resistant White Iron	Alloy A14 is a high chromium white cast iron offering high impact resistance and moderate erosion wear resistance. Alloy A14 is suitable for gravel pump applications where large slurry particles are present. A14 is much tougher than A05 but also exhibits a lower erosion wear resistance.
A25	Ni – Cr – Mo Steel	Cast Steel	Alloy A25 is an alloy steel having moderate wear resistance and high mechanical properties. The alloy is used for large castings where toughness is of primary importance.
A49	ULTRACHOME 28% Cr, Low C	Low Carbon, High Chromium White Iron	Alloy A49 is a corrosion resistant white iron suitable for low pH corrosion duties, where erosive wear is also a problem. The alloy is particularly suitable for Flu Gas Desulphurisation (FGD) and other corrosive applications, where the pH is less than 4. The alloy can also be used in other mildly acidic environments. A49 has an erosion resistance similar to that of Ni-Hard 1.
A51	ULTRACHOME 36% Cr, Low C	Erosion/ Corrosion White Iron	ULTRACHOME A51 is a premium erosion/corrosion alloy to be used where excellent erosion and corrosion resistance is required. The alloy has much improved corrosion resistance compared to alloy A49, whilst the erosion resistance is similar to Ni-Hard type alloy irons. The alloy is suitable for phosphoric acid duties, FGD duties, sulphuric acid , and other moderately corrosive applications.
C02	Ni-Resist (Copper free)	Corrosion Resistant Cast	Alloy C02 is a high nickel cast iron useful for light chemical duties involving low

WARMAN CODE	MATERIAL NAME	TYPE	DESCRIPTION
		Iron	concentration of solids.
C14	27 Cr-0.4 C5 Stainless Steel	Duplex Stainless Steel	Alloy C14 is a corrosion resistant stainless steel suitable for use in acidic environments. The alloy is particularly suitable for Flue Gas Desulphurisation (FGD) applications, where the pH is between 3 and 7. The alloy offers moderate erosion-corrosion resistance.
C21	Type 420C Stainless Steel	Martensitic Stainless Steel	Alloy C21 is a martensitic stainless steel having a combination of high hardness and good general corrosion resistance. The alloy is machined in the annealed, or as cast condition and is subsequently hardened for service.
C23	Type 316 Stainless Steel	Austenitic Stainless Steel	Alloy 23 (316SS) is an austenitic stainless steel having excellent corrosion resistance in reducing media. The molybdenum present in C23 increases its resistance to pitting corrosion. The alloy has good mechanical properties, however its low hardness gives it a low erosion resistance.
C25	Alloy 20	Austenitic Stainless Steel	Alloy C25 was specifically developed for sulphuric acid applications. The alloy can be used successfully in up to 85% Sulphuric acid. Alloy C25 also offers excellent corrosion resistance to a wide range of acids, and some strong alkalies. The alloy has poor resistance to erosive wear.
C26	26 Cr 5 Ni Stainless Steel (CD-4M Cu)	Duplex Stainless Steel	Alloy C26 is a corrosion resistant stainless steel suitable for use in acidic environments. The alloy offers moderate erosion-corrosion resistance.
C27	‘825’ Alloy	Austenitic Corrosion Resistant Alloy	Alloy C27 is an austenitic corrosion resistant alloy suitable for strong acid duties.
C30	27 Cr 31 Ni Stainless Steel	Austenitic Stainless Steel	Alloy C30 is an all purpose austenitic stainless alloy for service in high corrosive conditions. C30 has excellent resistance to general corrosion, pitting, crevice corrosion, intergranular corrosion and stress corrosion cracking. The alloy was developed originally for use in phosphoric acid.
C44	Type 440C Stainless Steel	Hardenable Stainless Steel	C44 is a martensitic stainless steel having a higher carbon level than 420C (C21) alloy. The higher carbon level results in

WARMAN CODE	MATERIAL NAME	TYPE	DESCRIPTION
			the formation of carbides within the microstructure. These carbides give 440C an improved wear life under the conditions of abrasive wear, as experienced with shaft sleeves. The formation of the carbide results in a reduction of the corrosion resistance of C44, compared with C21.
C55	Ferralium 255	Duplex Austenitic/ Ferritic Stainless Steel	Alloy C55 is a duplex ferritic – austenitic stainless steel. It combines high strength and physical properties with excellent corrosion resistance. Alloy C55 offers improved resistance to stress corrosion cracking, pitting and crevice corrosion over C22, C23 and C25 grades of stainless steel.
D21	Ductile Grey Iron (SG Iron)	Cast Iron	Alloy D21 is a ductile grade of grey iron used where higher physical properties and greater shock resistance are required compared to alloy G01
D81	Zinc Plated D21	Zinc Plated SG Iron	Alloy D81 is a zinc plated ductile iron which is used for duties where higher physical properties and greater shock resistance are required in comparison to G01. D21 has a better atmospheric corrosion resistance than D21.
G01	Grey Iron	Cast Iron	Alloy G01 is an inexpensive alloy used where high physical strength and erosion resistance are not required.
J21	Tungsten Carbide V21 coated C21	Ceramic Coated Stainless Steel	J21 is a ceramic coating (V21) applied over a C21 substrate. The combination of these two materials provides high abrasive wear resistance together with high toughness. The tungsten carbide layer is deposited onto the C21 substrate using a special spray technique which yields minimal porosity and excellent interlayer adhesion. J21 is unaffected by differential thermal expansion and will not “spall”.
J24	Tungsten Carbide V21 coated C23	Ceramic Coated Austenitic Stainless Steel	J24 consists of a V21 ceramic coating deposited onto a C23 substrate using a special spray technique. The coating is very hard and offers excellent abrasive wear resistance. The spray technique gives a coating with minimal porosity and excellent interlayer bond strength. J24 is unaffected by differential thermal expansion and will not “space”.
J25	Tungsten Carbide	Tungsten	J25 consists of V21 ceramic coating

WARMAN CODE	MATERIAL NAME	TYPE	DESCRIPTION
	V21 Coated C11	Carbide V21 Coated C11	deposited onto a C21 substrate using a special spray technique. The coating is very hard and offers excellent abrasive wear resistance. The spray technique gives a coating with minimal porosity and excellent interlayer bond strength. J25 is unaffected by differential thermal expansion and will not “spall”.
J26	Chrome Oxide (Y03) coated C26	Ceramic Coated Stainless Steel	J26 consists of Y03 Ceramic Coating deposited onto a C26 substrate using a special spray technique. The coating is very hard and offers excellent abrasive wear resistance. The spray technique gives a coating with minimal porosity and excellent interlayer bond strength. J26 is unaffected by differential thermal expansion and will not spall.
J27	WC/Chromium/ Nickel Coated C26	Tungsten Carbide V23 Coated C26	J27 consists of a WC/Cr/Ni (V23) coating deposited onto a duplex stainless steel (C26) substrate using a thermal spray technique. The coating offers both abrasive wear resistance and corrosion resistance exhibiting minimal porosity.
N02	63 Ni 30 Cu Alloy	Corrosion Resistant Alloy	Alloy N02 is a nickel based corrosion resistant alloy for use in reducing acids and chlorides. It is used extensively in pickling and marine applications.
N04	58 Ni 16 Cr 16 Mo Alloy	Corrosion Resistant Alloy	Alloy N04 is a nickel based corrosion resistant alloy specially resistant to oxidising acids and reducing chlorides based solutions.
N05	55 Ni 22 Mo Alloy	Corrosion Resistant Alloy	Alloy N05 is a chemical resistant alloy which can be used in non-oxidising environments. It has high physical properties and can be used successfully in high temperature environments.
N22	55 Ni 22 Cr 13 Mo Alloy	Corrosion Resistant Alloy	Alloy N22 is a nickel based corrosion resistant alloy specially resistant to extreme oxidising acids and reducing chloride based solutions. Its resistance to pitting in these solutions is superior to that of N04 and N05.
N23	55 Ni 22 Cr 13 Mo Alloy (Wrought)	Hastelloy® C22	Alloy N23 is a nickel based corrosion resistant alloy specially resistant to extreme oxidising acids and reducing chloride based solutions. Its resistance to pitting in these solutions is superior to that of N04 and N05.

WARMAN CODE	MATERIAL NAME	TYPE	DESCRIPTION
P09	Polyester Fibreglass DMC	Reinforced Structural Polymer	P09 is a reinforced polyester resin used for structural pump parts as a replacement for heavier grey and ductile iron parts. The combination of glass fibres and a crystalline resin provides a material with excellent mechanical properties.
P50	Polyphenylene Sulphide (Ryton*) * Ryton is a trade name of the Phillips Chemical Company)	Reinforced Structural Polymer	P50 is a high-strength plastic suitable for parts requiring high-dimensional stability.
P60	UHMW Polyethylene	Engineering Polymer	
R08	Standard Impeller Rubber	Natural Rubber	R08 is a black natural rubber, of low to medium hardness. R08 is used for impellers where superior erosive resistance is required in fine particle slurries. The hardness of R08 makes it more resistant to both chunking wear and dilation (ie, expansion caused by centrifugal forces) as compared to R26. R08 is generally only used for impellers.
R24	Anti Thermal Breakdown Rubber	Natural Rubber	Anti Thermal Breakdown Rubber (ATB) is a soft natural rubber based on R26, but with improved thermal conductivity. It is intended for use as a liner material in slurry pumping applications where high impeller peripheral speeds are required.
R26	Standard Liner Rubber	Natural Rubber	R26 is a black, soft natural rubber. It has superior erosion resistance to all other materials in fine particle slurry applications. The antioxidants and antidegradents used in R26 have been optimised to improve storage life and reduce degradation during use. The high erosion resistance of R26 is provided by the combination of its high resilience, high tensile strength and low short hardness.
R33	Natural Rubber – Reinforced	Natural Rubber	R33 is a premium grade material for use where R26 does not provide sufficient wear life. It is a black natural rubber, of low hardness and is used for cyclone and pump liners and impellers where its superior physical properties give increased cut resistance to hard, sharp slurries.

WARMAN CODE	MATERIAL NAME	TYPE	DESCRIPTION
R38	Natural Rubber Reinforced	Natural Rubber	R38 is a black natural rubber, of medium hardness. R38 is used for impellers where superior erosive and tear resistance is required in fine particle slurries. The hardness and tear resistance of R38 makes it more resistant to both chunking wear and dilation (ie, expansion caused by centrifugal forces) as compared to R26 and R08. R38 is generally only used for impellers.
R66	60 Duro Natural Rubber	Natural Rubber	This is a hard (60 Duro) natural rubber product used for FGD duties primarily in GSL Pumps.
S01	EPDM Elastomer	Synthetic Elastomer	S01 is an acid and ozone resistant rubber which is of low abrasion resistance. EPDM is non polar and difficult to bond to metal, therefore it is used typically in lipseals and volute seal applications.
S02	EPDM General Rubber	Synthetic Elastomer	S02 is an acid and ozone resistant rubber which is of medium abrasion resistance. EPDM is non polar giving it special chemical resistance. S02 is a speciality elastomer for use only in applications that require the properties of EPDM.
S03	High Temperature EPDM	EPDM	S03 is a high temperature and chemical resistant EPDM elastomer. It has been compounded so as to have a very low compression set and is therefore designed for use in sealing applications. This material is not designed for general use in parts subjects to erosive wear.
S12	Nitrile Rubber	Synthetic Elastomer	Elastomer S12 is a synthetic rubber which is generally used in applications involving fats, oils and waxes. S12 has moderate erosion resistance.
S21	Butyl Rubber	Synthetic Elastomer	Butyl rubber is a highly saturated elastomer which has excellent chemical stability, and good resistance to heat and oxidation. The high saturation reduces the elastomeric properties of S21, and hence reduces its erosion resistance. In general S21 is used in acidic environments.
S31	Chlorosulfonated Polyethylene (Hypalon*) * Hypalon is a trademark of the	Synthetic Elastomer (CSM)	S31 is an oxidation and heat resistant Elastomer. It has a good balance of chemical resistance to both acids and hydrocarbons.

WARMAN CODE	MATERIAL NAME	TYPE	DESCRIPTION
	Dupont Company		
S42	Polychloroprene (Neoprene*) * Neoprene is the trademark of the Dupont Company	Synthetic Elastomer (CR)	Polychloroprene (Neoprene) is a high strength synthetic elastomer with dynamic properties only slightly inferior to natural rubber. It is less effected by temperature than natural rubber, and has excellent weathering and ozone resistance. It also exhibits excellent oil resistance.
S45	High Temperature Hydrocarbon Resistant Rubber	Synthetic Elastomer	S45 is an erosion resistant synthetic rubber with excellent chemical resistance to hydrocarbons at elevated temperatures.
S51	Fluoroelastomer (Viton*) * Viton is the trademark of the Supon Company	Synthetic Elastomer (FPM)	S51 has exceptional resistance to oils and chemicals at elevated temperatures. Limited erosion resistance.
U01	Wear Resistant Polyurethane	Polyurethane Elastomer	U01 is an erosion resistant material that performs well in elastomer applications where 'tramp' is a problem. This is attributed to the high tear and tensile strength of U01. However, its general erosion resistance is inferior to that of natural rubber (R26, R08).
Y07	Alumina 99%	Ceramic	Wear resistant ceramic.
Y08	Silicon Nitride Bonded Silicon Carbide	Ceramic	Wear resistant ceramic.
Y11	Fine Grained SiN/SiC	Wear Resistant Ceramic	Y11 is produced by bonding a fine grained silicon carbide powder with silicon nitride. The ceramic has high thermal shock resistance and physical properties. Y11 has a high wear resistance, compressive strength and modulus of rupture than Y08.
Y14	Reaction Bonded Silicon Carbide	High Wear Resistant Ceramic	Y14 is produced by reaction bonding silicon carbide grains with silicon nitride. The ceramic has high thermal shock resistance and physical properties, high wear resistance and high corrosion resistance. Y14 can be manufactured in thin sections of 5mm up to approximately 25mm.
Z11	Material Composite Y11/U01	Ceramic Polyurethane Combination	Z11 is a useful material for applications requiring low angle erosion and abrasion resistance. The Y11 Silicon Nitride bonded Silicon Carbide tiles provide a

WARMAN CODE	MATERIAL NAME	TYPE	DESCRIPTION
			very hard wear resistant surface with the U01 polyurethane providing support. The polyurethane backing allow the 'brittle' ceramic tile to float and absorb higher angle and large particle impacts.
Z12	Material Composite Y11/A12	Ceramic Alloy Combination	Z12 is the combination of Y11 Nitride bonded Silicon Carbide and A12 Ultrachrome 27% Cr White Iron. It is application for parts that require resistance to low angle erosion and sliding abrasion for particle sizes up to 5mm.
Z13	Material Composite Y11/A05	Ceramic Alloy Combination	Z13 is the combination of Y11 Nitrided bonded Silicon Carbide and A05 Ultrachrome 27% Cr White Iron. It is applicable for parts that require resistance to low angle erosion and sliding abrasion for particle sizes up to 5mm.
Z14	Reaction Bonded Silicon Carbide/Foam	Ceramic/ Polyurethane Foam Combination	Z14 is used in cyclone spigot liners. The ceramic Y14 is coated in polyurethane foam. This foam provides protection and a light weight coating to seat the thin walled ceramic in position in the DMC casing.
Z15	Nitride Bonded Silicon Carbide / Polyurethane	Ceramic / Polyurethane Combination	Z15 is a useful material for applications requiring low angle erosion and abrasion resistance. The Y08 Nitride bonded Silicon Carbide tiles provide a very hard, wear resistance surface with the U01 polyurethane providing support. The polyurethane backing allows the 'brittle' ceramic tile to float and absorb higher angle and large particle impacts.
Z16	Nitride Bonded Silicon Carbide / Ultrachrome™ 27% Cr	Ceramic / White Iron Combination	Z16 is the combination of Y08 Nitrided bonded Silicon Carbide and A05 Ultrachrome™ 27% Cr White Iron. It is applicable for parts that require resistance to low angle erosion and sliding abrasion for particle sizes up to 1000μm.

APPENDIX 3 – SLURRY FRICTION HEAD LOSSES IN PIPELINES

INTRODUCTION

Despite the long history of successful slurry pumping operations covering a wide range of slurries, limited published data is available to estimate Hf accurately, for every possible duty. A high degree of accuracy is normally required only if Hf represents a high proportion of the Total Dynamic Head, H, for a proposed application so that large errors in estimating Hf would be reflected in correspondingly large errors in estimating H.

This normally applies to very long distance pumping duties only. For most Warman Pump applications, a high degree of accuracy in estimating Hf is not required.

HOMOGENEOUS SLURRIES: (PARTICLES ESSENTIALLY ALL FINER THAN 50μm)

At sufficiently low concentrations Hf will be close to that for clear water and may be estimated by the same empirical method as applied to Category 'A' Heterogeneous Slurries.

At sufficiently high concentrations, the Yield Stress characteristic largely influences the value of Hf. For further information on pumping, homogeneous slurries with high concentrations, contact your nearest Warman representative.

HETEROGENEOUS SLURRIES: CATEGORY 'A'

Category 'A': Particles essentially all coarser than 50μm and finer than 300μm and with Cw from ZERO to 40%.

Typical friction head loss curves for this category are illustrated in Figure A3-1. Analyses of Hf data on these slurries indicates that, for any given solids concentration, the slurry Hf is numerically higher than the water Hf, for velocities below approximately 1.30 VL. However, the Hf value does not fall below a minimum at lower velocities, due to the effect of solids which settle in the pipe. This minimum occurs at approximately 0.70VL, where the slurry Hf is approximately numerically equal to the Hf for water at VL.

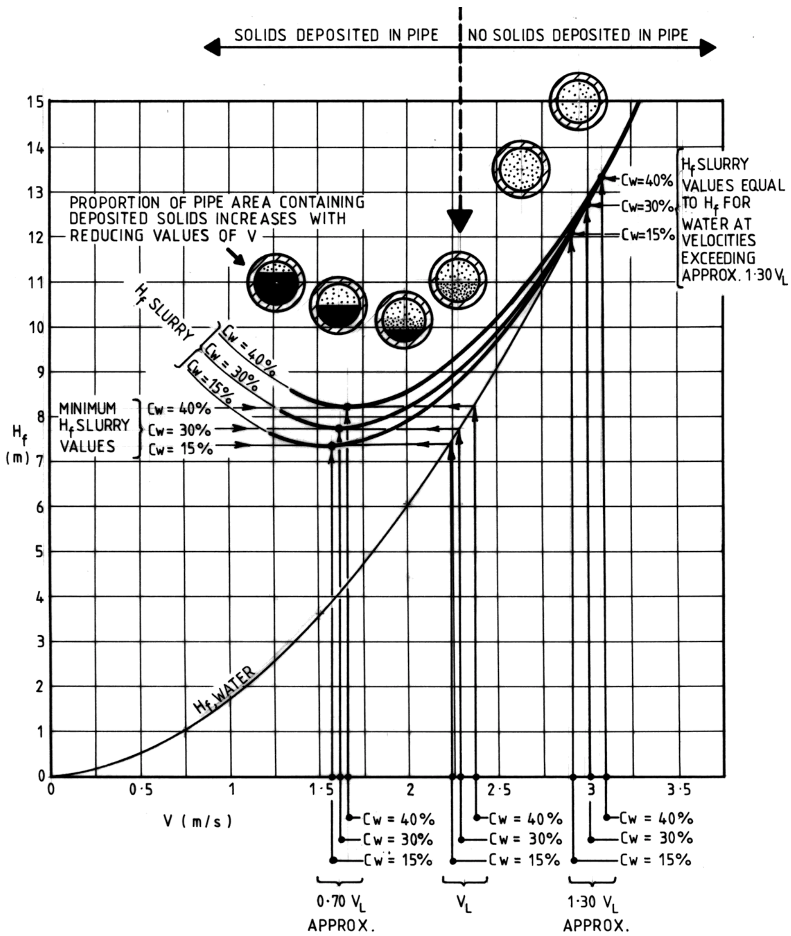


FIGURE A3-1 TYPICAL Hf CURVE FOR CATEGORY 'A' SLURRIES

The empirical data are summarised:

At 1.30 VL (approximately):

Slurry Hf is numerically equal to water Hf.

At 0.70 VL (approximately):

Slurry Hf is at its minimum value.

Slurry Hf is numerically equal to the Hf for water at VL.

The most economical slurry velocity is a velocity a little in excess of VL, thus these empirical relationships allow the construction of the useful portion of the estimated slurry Hf curve, in relation to the water Hf curve, for the same pipe.

Consequently, a reliable method of estimating Hf for water should be adopted, when estimating a Category 'A' Slurry Hf.

NOTE: Both water Hf (head of water) and slurry Hf (head of mixture) should each be expressed in head of actual "mixture" pumped.

Figure A3-1 also illustrates the construction of the estimation slurry Hf curves, based upon the estimated water Hf curve. Each slurry Hf curves meets tangentially with its minimum Hf value and meets tangentially with the water Hf curve where the value of VL corresponds to 1.30 VL.

It is emphasised that this empirical method of estimating Hf for these Category ‘A’ slurries is not precise but, in the absence of pipeline test rig data, or other more reliable data, it provides estimates considered to be reasonably accurate for many practical slurry pumping applications.

HETEROGENEOUS SLURRIES: CATEGORY ‘B’

Category ‘B’: Particles essentially all coarser than 50µm and finer than 300µm but with Cw greater than 40%.

Generally, friction head losses for this category are much higher than for Category ‘A’ due largely to the increased friction effect of the more closely-packed solids content upon the pipewall. This effect generally increases with increasing Cw and is so greatly influenced by a number of variables, for example, Cw, S, Sl, d50, and actual sieve analysis of solids present that it is not possible to provide a simple empirical method of estimating slurry Hf.

In general, slurry Hf values may vary over a range, commencing with values approximately equal to those applicable to Category ‘A’ slurries at Cw = 40%, to values up to double or more those of Category ‘A’ slurries, for velocities in excess of VL.

Consequently, Hf values for Category ‘B’ slurries must often be estimated, then adjusted by an “experience factor”. The Hf values are first estimated as if for category ‘A’, after allowing for the lower values of FL (and VL) associated with values of Cw in excess of Cw = 30%, see Figure A3-1.

The true values of Hf may be double or more the estimated values. This is allowed for by providing reserves of speed and power for values of Hf up to double, or more, of the values estimated for Hf. While this introduces the risk of large error in the estimation of Hf, the effective overall error in estimating Total Dynamic Head (H) is relatively small, if the other components of H (for example, Z, Hpf and Hve), when combined, represent the major portion of H.

Should the value of H be estimated with a relatively small error, the effect would probably be almost insignificant. For example, it would simply result in a slightly higher or lower value of Zs in the hopper and/or a correspondingly slightly higher or lower power consumption. Should the error be more significant, with obvious overspeed or underspeed, the pump speed may be adjusted, for example, by changing the motor pulley or via a variable speed control, if provided. In either case, the drive motor should be adequately rated.

NOTE: Some test work results for slurry containing heavy solids (S = 4.6 to 5.3) of approximately 150µm sizing has shown a trend towards decreasing head loss with increasing solids concentration, between Cv = 10% and Cv = 25% (that is, Cw between approximately 40% and 60%).

Many Warman pumps are used in heavy-duty Category 'B' slurry applications.

Typical examples includes the following:

a) Mill Discharge Plant;

b) Thickener Underflow;

c) Sand Tailings Stacking, and;

d) Gravity Concentrator Feed.

HETEROGENEOUS SLURRIES: CATEGORY 'C'

Category 'C': Particles essentially coarser than 300μm and Cw from ZERO to 20%.

Generally, friction head losses for Category 'C' slurries are also much higher than for Category 'A'.

The more common applications for Warman pumps on Category 'C' slurries are the suction dredging of gravel and/or coarse sand. In normal dredging operations, Cw is often less than 20%, due to the impracticality of continuously entraining such coarse particles at the intake of the suction pipe at a higher value of Cw.

Hf for these slurries is estimated on the basis of the minimum average velocity FOR DESIGN being no less than VL when FL = 1.4.

<u>For VL and for velocities greater than VL</u> – the slurry Hf is taken to be numerically equal to 1.10 Hf for water, that is, numerically 10% higher than the estimated water Hf.

HETEROGENEOUS SLURRIES: CATEGORY 'D'

Category 'D': Particles essentially coarser than 300μm and Cw greater than 20%.

Generally, friction head losses for Category 'D' slurries are higher than for Category 'A'. The values of Hf may be first estimated by the same method as for Category 'A'. However, the true slurry Hf may vary from values close to those for Category 'A' up to three times or more those of Category 'A' slurries, (for velocities in excess of VL). Consequently, reserves of speed and power should be provided.

ESTIMATION OF FRICTION HEAD LOSSES FOR CLEAR WATER

The recommended method for estimating Hf for clear water is by using Darcy's formula as follows:

$$Hf = f \times \frac{L}{D} \times \frac{V^2}{2g} \quad \text{where,}$$

Hf	=	*Friction Loss (m)*
L	=	*Total Equivalent Length of Pipe (m)*
D	=	*Inside diameter of pipe (m)*
F	=	*Darcy Friction Factor*
V	=	*Velocity (m/s)*
G	=	*Gravitational Acceleration (9.81 m/s^2)*

Use the Warman Pipe Friction Chart, Figure A3-2, to evaluate the Darcy Friction Factor, f.

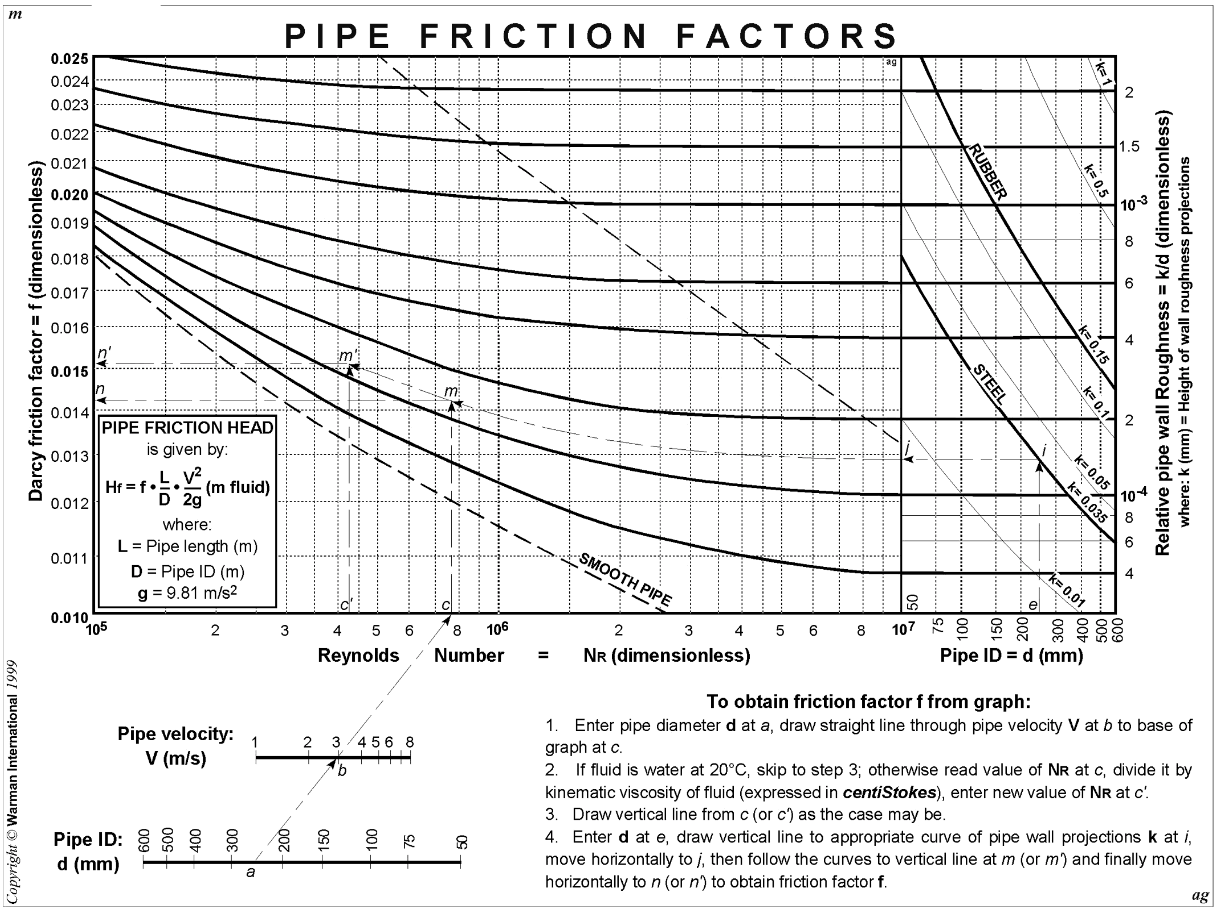


FIGURE A3-2 WARMAN PIPE FRICTION CHART

NOTE: For convenience, this chart is entered at values of Inside Diameter of Pipe: "d", expressed in mm.

a) The application of Darcy's formula, in combination with the Warman Pipe Friction Chart, is the recommended method of estimating Hf for water. This information should then be used for construction of the System Resistance

Curves for clear water and Category 'A' slurries (by the empirical method) illustrated in Figure A3-1.

The advantages of this procedure are:

i. The Warman Pipe Friction chart provides the Darcy Friction Factor (and thus Hf) values for clear water based on the most reliable data available to the date of this publication. This data take into account the maintenance of certain values for Relative Pipe Wall Roughness, k/d, due to the continuous 'polishing' action of abrasive slurries flowing through the pipes.

   For example, on Figure A3-2, the values of k/d for 'commercial steel' pipes are the same as the values for 'cement' and 'polythene' pipes. However, when these pipes are used for handling non-abrasive liquids only, such as clear water, the true values of k/d for steel pipes would be actually a little higher, yielding correspondingly higher values of Hf for water.

ii. The empirical method for the construction of the estimated System Resistance Curve for water, and the subsequent construction of the System Resistance Curve for slurry allows for the varying degrees of difference between Hf for water and Hf for slurries. This is particularly the case in the range of flow rate between VL to 1.30 VL, which is the usual range of most interest.

b) Example of Friction Head Loss Estimation for Water

Given L = *700m of commercial steel pipe*

d = *200mm (ie: D = 0.200m), see Figure A3-3*

Q = *94.25 L/s*

g = *9.81 m/s²*

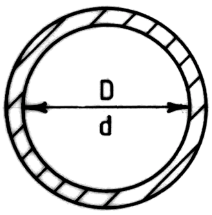


FIGURE A3-3 PIPE DIAMETER

c) Calculations

Cross-sectional areas of Pipe at inside Diameter

$$= \frac{D^2}{4}\pi = \frac{0.2^2}{4}\pi \; x \; 3.1416 = 0.031416 m^2$$

As 1 m^3 = *1000 litres,*

$$Flow\ rate\ in\ m^3/s\ =\ \frac{Q}{1000}\ =\ \frac{94.25}{1000}\ =\ 0.09425\ m^3/s$$

$$As\ V\ =\ \frac{Flow\ rate\ (m^3/s)}{Cross\ sectional\ area\ of\ \ pipe\ (m^2)}$$

$$Average\ Pipeline\ Velocity,\ V\ =\ \frac{0.0942}{0.031416}\ =\ 3.0\ m/s$$

Alternatively,

$$V\ =\ \frac{1273\ x\ Q}{d^2}\ =\ \frac{1273\ x\ 94.25}{200\ x\ 200}\ =\ 3.0\ m/s$$

Refer to the Warman Pipe Friction Chart, Figure A3-2.

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**^{7}’ 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 a dimensionless 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, Hf can be evaluated as follows:

$$Hf\ =\ f\ \ x\ \frac{L}{D}\ x\ \frac{V^2}{2g}\ \ (m)$$

$$Hf\ =\ 0.0158\ \ x\ \frac{700}{0.200}\ \ x\ \frac{3.0^2}{2\ x\ 9.81}\ \ =\ 25.4m$$

APPENDIX 4 – TOTAL DYNAMIC HEAD

ABSTRACT

The main components of Total Dynamic Head are:

a) Total Discharge Head, and

b) Total Suction Head.

The equation is:

Total Dynamic Head = Total Discharge Head - Total Suction Head

Algebraically, $H = (Hd) - (Hs)$ *(m)*

or, $H = (Hgd + Hvd) - (Hgs + Hvs)$ *(m)*

The values Hvd and Hvs are always positive (+ve)

Hd is usually positive (+ve), (above pump centreline)

Hs may be positive (+ve), (above pump centreline) or negative (-ve), (below pump centreline).

When Hs is positive (+ve): $H = (Hd) - (Hs)$ *ie:* $H = Hd - Hs$

When Hs is negative (-ve): $H = (Hd) - (Hs)$ *ie:* $H = Hd + Hs$

a) Total Discharge Head, Hd

Basic Simple Formula: $Hd = Zd + Hfd + Hve$ *(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 Hd, 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, Hs

Basic Simple Formula: $(Hs) = (Zs) - Hi - Hfs$ *(m)*

(Hs) 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 Hs due to any contractions, enlargements, flow measuring device. These are as follows:

i) liquid supply surface being under pressure, Hpr, or under vacuum, Hvac;

ii) differential column head loss, Zc, and

iii) substitution of effective mixture static suction head Zsm in lieu of Zs.

NOTE: Values of Hs are directly applicable in NPSHa calculations and in selection of shaft-sealing arrangements.

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.

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 A4-1 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.

$$H = Hd - (Hs)$$

$$= (Hgd + Hvd) - (Hgs + Hvs)$$

$$\text{where } Hvd = \frac{Vd^2}{2g} \text{ and } Hvs = \frac{Vs^2}{2g}$$

These velocities represent the actual values for average velocity at the pump discharge flange (Vd), and at the pump suction flange (Vs), respectively.

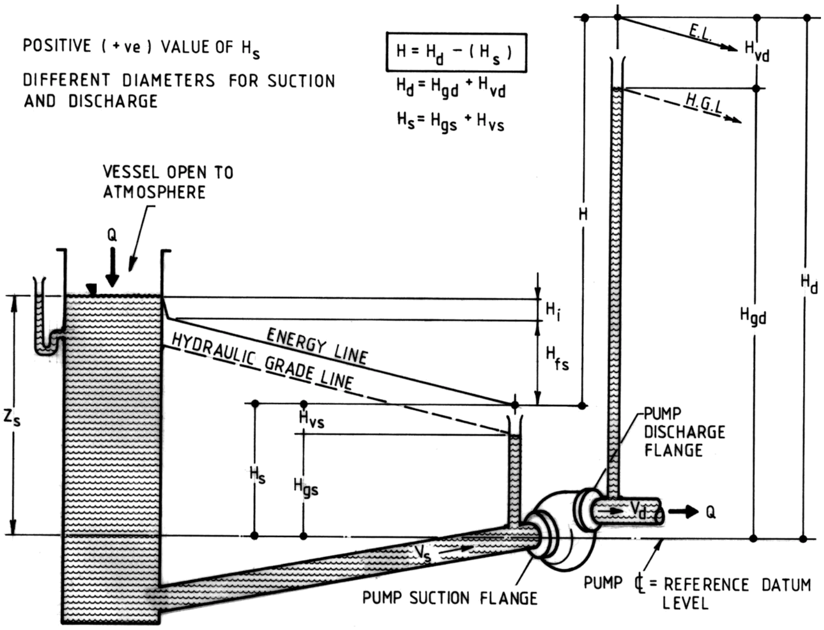


FIGURE A4-1 TOTAL DYNAMIC HEAD WITH POSITIVE INTAKE HEAD

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

When Hs is negative (-ve) that is, a vacuum head is indicated by the gauge, as in Figure A4-2, the substitution of the negative value in the formula serves to positively increase the value of H with respect to Hd.

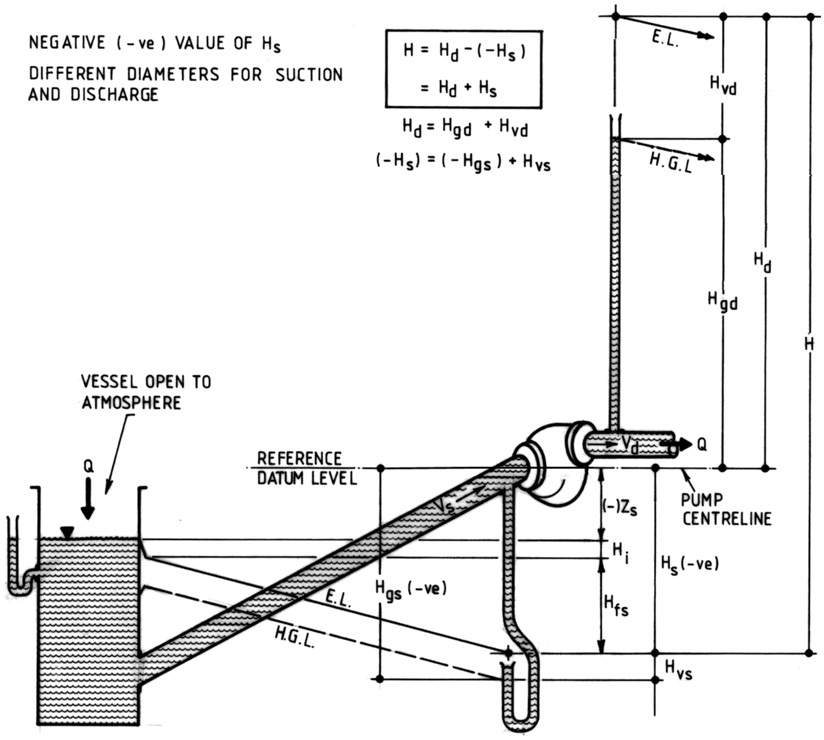


FIGURE A4-2 TOTAL DYNAMIC HEAD WITH NEGATIVE INTAKE HEAD

ESTIMATION OF TOTAL DYNAMIC HEAD

As H = (Hd) – (Hs) and as the suction and discharge pipes are often of different internal diameter, it is advisable to estimate values of Hd and Hs 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: Hd

Basic Simple Formula: Hd = (Zd) + Hfd + Hve

Typical possible additional terms are as follows:

i) Head Loss on conical enlargement (see Figure A4-4);

ii) Head Loss on contraction (see Figure A4-4);

iii) Head Loss on Exit into Pressure-Fed Equipment (refer to **Head Loss at Exit into Pressure-Fed Equipment**).

b) Total Discharge Head: Hs

Basic Simple Formula: Hs = (Zs) + Hi + Hfs

Typical possible additional terms are as follows:

i) Head GAIN in supply from Pressure Vessel, Hpr;

ii) Head Loss in supply from Vacuum Vessel, Hvac

iii) Head Loss on Differential Column (applicable in dredge applications, Zc – refer to **Differential Column Head Loss**)

SEPARATE ESTIMATES OF SUCTION HEAD & DISCHARGE HEAD

a) Pipeline Friction Head Loss, Hf

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, La(m) and the Aggregate of Equivalent Lengths for all valves, bends and like fittings, Lf(m) (see Figure A4-3) contributing to friction head loss in the pipeline.

	Radius More Than 3 x N.B.	Radius is 2 x N B.			Minimum Radius 10 x N.B.				
INTERNAL DIAMETER or N.B.	90° Long Radius Bend	90° Short Radius Bend	Elbow	Tee	Rubber Hose	Diaphragm Valve Full Open	Full Bore Valve Round Way	Plug-Lub Valve Rect. Way	*"Tech-Taylor" Valve Ball Type
mm	EQUIV. LENGTH IN m OF STRAIGHT PIPE GIVING EQUIVALENT RESISTANCE TO FLOW								
25	0·52	0·70	0·82	1·77	0·30	2·56	——	0·37	——
32	0·73	0·91	1·13	2·38	0·40	3·29	——	0·49	——
40	0·85	1·10	1·31	2·74	0·49	3·44	1·19	0·58	——
50	1·07	1·40	1·68	3·35	0·55	3·66	1·43	0·73	——
65	1·28	1·65	1·98	4·27	0·70	4·60	1·52	0·85	——
80	1·55	2·07	2·47	5·18	0·85	4·88	1·92	1·04	0·20
90	1·83	2·44	2·90	5·79	1·01	——	——	1·22	——
100	2·13	2·77	3·35	6·71	1·16	7·62	2·19	1·40	0·23
115	2·41	3·05	3·66	7·32	1·28	——	——	1·58	——
125	2·71	3·66	4·27	8·23	1·43	13·11	3·05	1·77	0·30
150	3·35	4·27	4·88	10·06	1·55	18·29	3·11	2·13	0·37
200	4·27	5·49	6·40	13·11	2·41	19·81	7·92	2·74	0·82
250	5·18	6·71	7·92	17·07	2·99	21·34	10·67	3·47	0·61
300	6·10	7·92	9·75	20·12	3·35	28·96	15·85	4·08	0·76
350	7·01	9·45	10·97	23·16	4·27	28·96	——	4·88	0·91
400	8·23	10·67	12·80	26·52	4·88	——	——	5·49	1·04
450	9·14	12·19	14·02	30·48	5·49	——	——	6·22	1·16
500	10·36	13·11	15·85	33·53	6·10	——	——	7·32	1·25

*"TECH-TAYLOR" VALVE IS A BALL TYPE CHANGEOVER DEVICE USED ONLY ON THE DELIVERY SIDE OF THE PUMP

NOTE: 1. FOR 135° BEND, USE 50% OF EQUIVALENT LENGTH FOR 90° BEND.

2. L_f IS THE AGGREGATE OF EQUIVALENT LENGTHS FOR ALL PIPELINE FITTINGS AND VALVES IN A GIVEN PIPELINE.

FIGURE A4-3 EQUIVALENT LENGTHS OF PIPE FITTINGS AND VALVES

Generally L = La + Lf.

Specifically:

i) For Suction Side: Ls = Las + Lfs (Friction Head Loss = Hfs)

ii) For Discharge Side: Ld = Lad + Lfd (Friction Head Loss = Hfd)

Values of Hfs and Hfd should be estimated separately, for example, during the preparation of the respective separate sets of calculations leading to the estimates of Hs and Hd. By separately estimating Hs, its value is readily available for use in NPSHa CALCULATIONS, (refer to APPENDIX 6 – NET POSITIVE SUCTION HEAD (NPSH)), and in the selection of Shaft-Sealing arrangements (refer to **Shaft Sealing**).

b) Inlet Head Loss, Hi: Exit Velocity Head Loss, Hve

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

i) Hi, the Inlet Head Loss (Suction side only), and

ii) Hve, the Exit Velocity Head Loss (Discharge side only).

That is, the terms Hi and Hve are included in the standard formulae for Hs and Hd respectively.

c) Head Losses due to Contractions and Enlargements

These additional head losses are calculated by use of the formulae provided in Figure A4-4. As no separate provisions are made in the standard Hs and Hd 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 Hfs or Hfd respectively.

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

GROUPS 1 TO 5 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 UPSTREAM VELOCITY AND V_1 THE DOWNSTREAM VELOCITY.

GROUP	ITEM	HEAD LOSS (m)
1.	Loss of head at inlet Hi From pump hopper to pump or from storage tank to pump.	
	(a) Flush Connections.	$0.5 \frac{V_1^2}{2g}$
	(b) Projecting connection and dredge suction pipes.	$1.0 \frac{V_1^2}{2g}$
	(c) Rounded Connection.	$0.05 \frac{V_1^2}{2g}$
2 θ	Loss of head due to conical enlargement from pump discharge flange to discharge pipeline included angle θ: 6° / 65° factor K_e: 0.14 / 1.15 *	$K_e \frac{(V-V_1)^2}{2g}$

GROUP	ITEM	HEAD LOSS (m)
3. d_1 d_2	Loss of head due to sudden contraction: Kc is a factor depending on ratio $\frac{d_1}{d_2}$ where d_1 is the large diameter and d_2 the small diameter as illustrated.	$K_c \frac{V_1^2}{2g}$
4 $V \rightarrow V_1$	Loss of head due to sudden enlargement:	$\frac{(V-V_1)^2}{2g}$
5	loss of head due to conical contraction : e.g. Jet Nozzles SEE CAMERON PAGE 3 – 110	$K_g \frac{(V-V_1)^2}{2g}$

Ratio d_1/d_2	1.2	1.4	1.6	1.8	2.0	2.5	3.0	4.0	5.0
Factor Kc	0.08	0.17	0.26	0.34	0.37	0.41	0.43	0.45	0.46

* FOR CONICAL ENLARGEMENTS, MAXIMUM HEAD LOSS OCCURS WHEN INCLUDED ANGLE IS 65°, WHEN $K_e = 1.15$. MINIMUM HEAD LOSS OCCURS WHEN INCLUDED ANGLE IS 6°, WHEN $K_e = 0.14$.

FIGURE A4-4 HEAD LOSSES AT INLET, CONTRACTION AND ENLARGEMENT

d) Sundry Additional Causes of Effects on Hfs or Hfd

The calculated values for Hfs and Hfd 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 A4-5 depicts a mixture of Specific Gravity, Sm, flowing upwards and drawn from a supply of settled solids and overlying liquid, S*l*. As the liquid of the same vertical height, Z*l*. The resulting effective static head loss is known as the differential Column Head loss, Zc:

$$where\ Zc = Zl\left(\frac{Sm - Sl}{Sm}\right)$$

(m) Sm 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

Where this condition exists, Zc must be included as an additional head loss in the pipe system. This would effect both total head and NPSHa (refer to APPENDIX 6 – NET POSITIVE SUCTION HEAD (NPSH)).

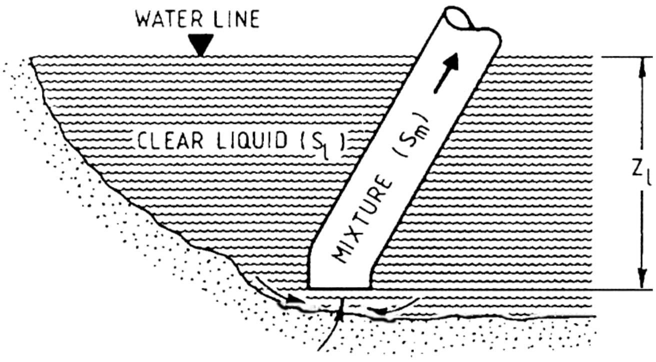


FIGURE A4-5 DIFFERENTIAL COLUMN HEAD LOSS

APPENDIX 5 – LIMITING SETTLING VELOCITY

GENERAL NOTES

Slurries containing essentially fine particles (predominantly less than 50 microns (0.05mm) 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 predominantly 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 VL. At any velocity below VL, 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.

DETERMINATION OF LIMITING SETTLING VELOCITY

In order to determine VL 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 VL may be established by a skilled specialist or estimated by one of the following methods, each based on Durand’s formula:

Durand’s Formula

$$VL = FL\sqrt{2gD \frac{(S - Sl)}{Sl}}$$

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 <u>closely – graded particle sizing</u> (see Figure A5-1).

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.

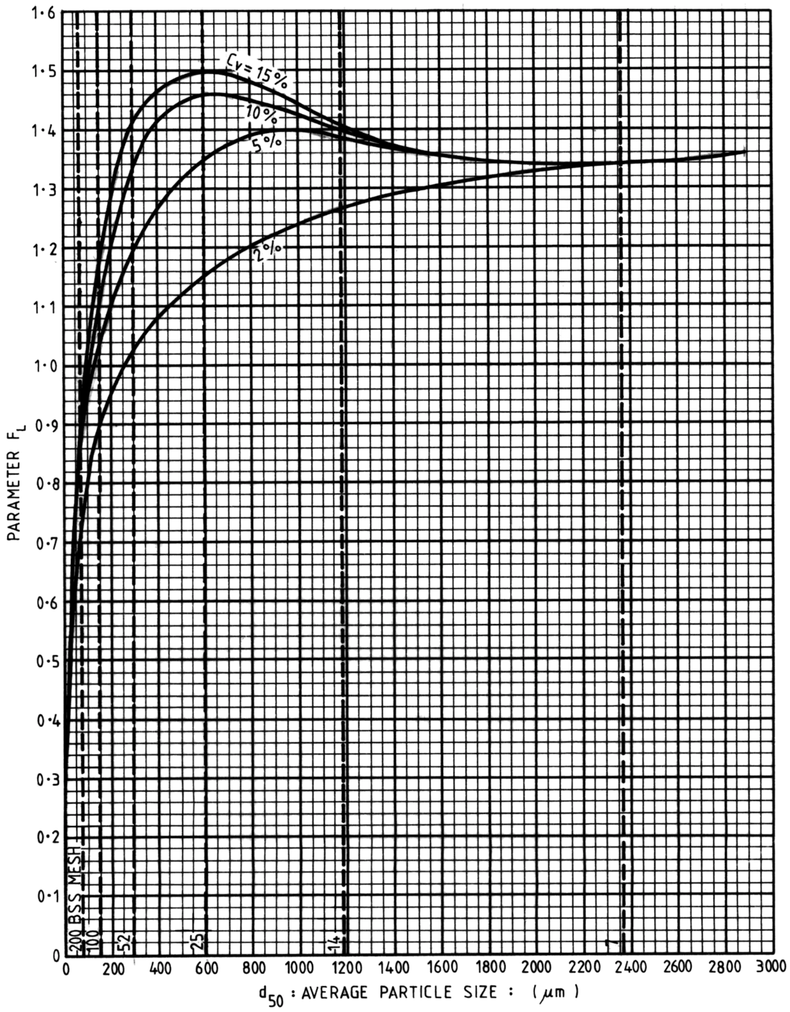


FIGURE A5-1 DURAND'S LIMITING SETTLING VELOCITY PARAMETER
(For particles of closely graded sizing)

Subsequent tests indicate that values of FL, (from Figure A5-1), provide conservative (high) values for VL in respect of:

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

b) Slurries of sizing containing <u>significant proportions of particles finer than 100µm</u>.

It is important that values of FL (and VL) are not excessively conservative (high). Excessively conservative estimates of FL (and VL) 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 the values for d50 and Cv; values of FL are obtained from Figure A5-1.

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

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

Figure A5-2 represents the results of field tests on slurries of widely-graded sizing. The particle sizing is simply expressed by the d50 term.

The resultant values of FL (and consequently, VL) are significantly below those which would be yielded from Figure A5-1.

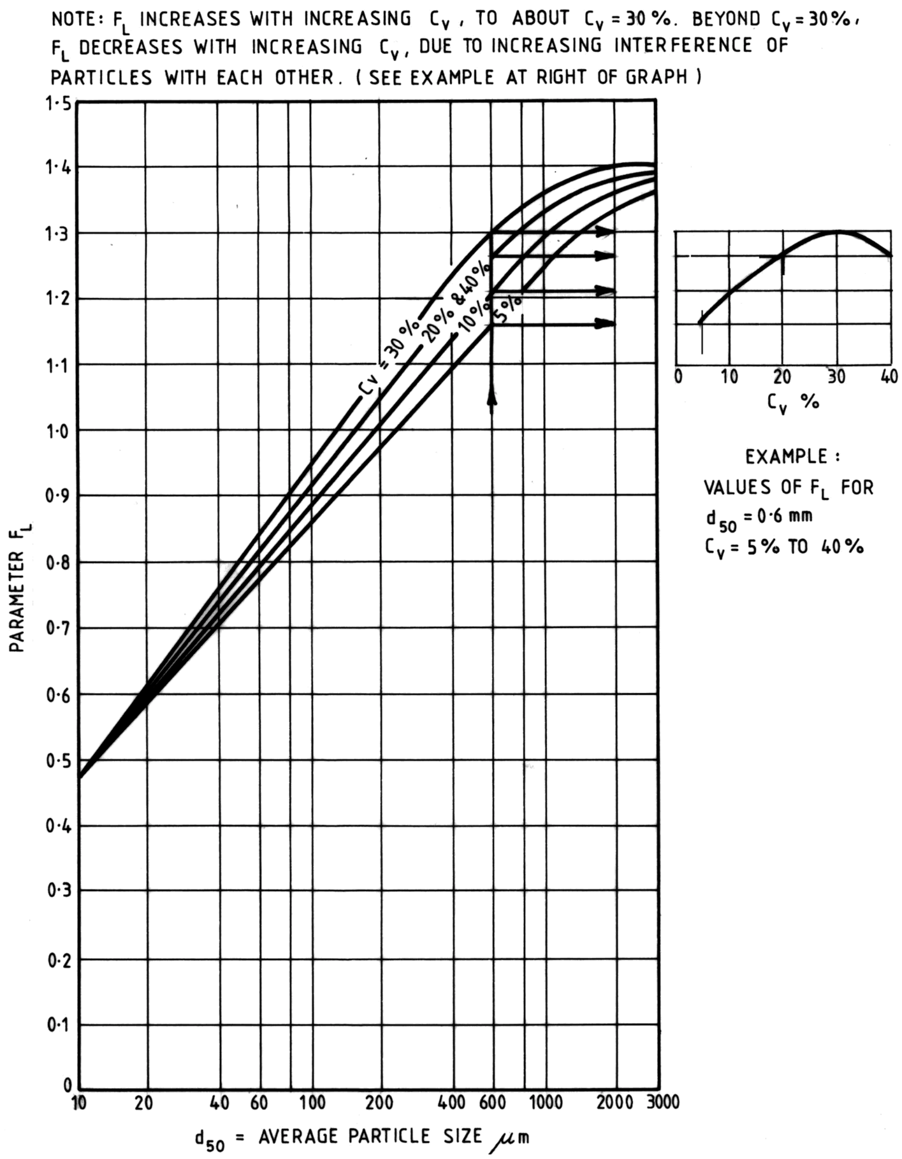


FIGURE A5-2 MODIFIED DURAND'S LIMITING SETTLING VELOCITY PARAMETER
(For particles of widely graded sizing)

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.

APPENDIX 6 – NET POSITIVE SUCTION HEAD (NPSH)

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 cracking 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 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 Net Positive Head at that point is equal to the gauge head reading, plus atmospheric pressure head, minus the liquid vapour pressure head, plus the velocity head. Gauge readings above atmospheric are taken as positive and below atmospheric as negative.

The Net Positive Head at the suction inlet of a pump is called the NPSH and the minimum NPSH required to avoid cavitation is usually shown on pump performance curves as "NPSH required" (NPSHr).

b) NPSH Available

For a particular pump installation, the NPSH available must be determined from the system conditions and liquid characteristics. The NPSH available (NPSHa) must exceed the NPSH required by the pump at the duty point, to prevent cavitation from occurring.

FORMULAE FOR NPSHA

Formulae for calculating NPSHa are shown in Figure A6-1a, b, c and d. All formulas refer to solids free, Newtonian liquid.

The diagrams are schematic only. They are used to clarify symbols and do not necessarily represent the best installation practice.

NOTES:

i. Use equation (1) to calculate NPSHa from pump tests.

ii. Use equation (2) to predict NPSHa from the installation drawings and design data.

iii. Express all the heads as liquid columns of density corresponding to the pumping temperature.

iv. Correct the barometric pressure (Hatm) for altitude (Figure A6-2).

v. Typical vapour pressures for water are given in Figure A6-3.

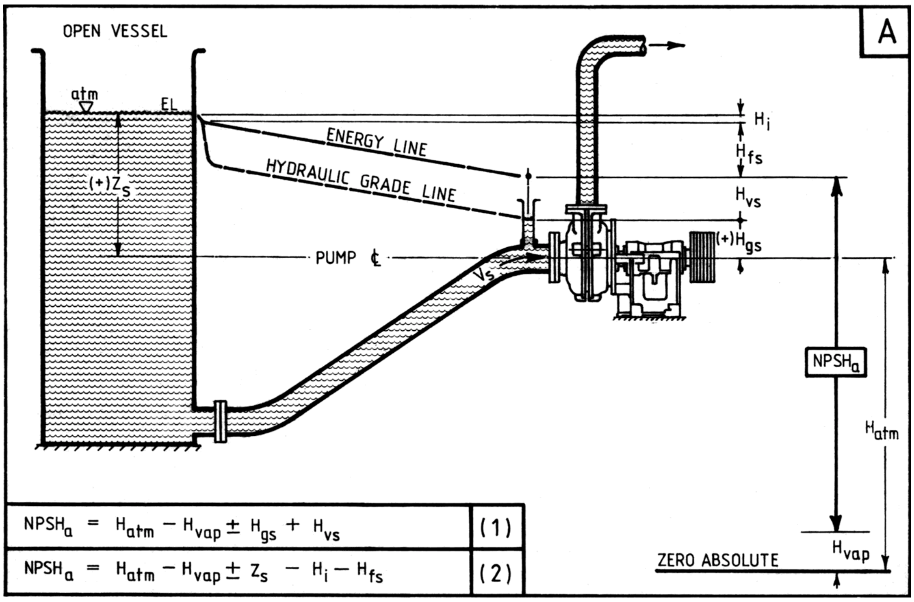


FIGURE A6-1a NPSHa FOR POSITIVE SUCTION CONDITIONS

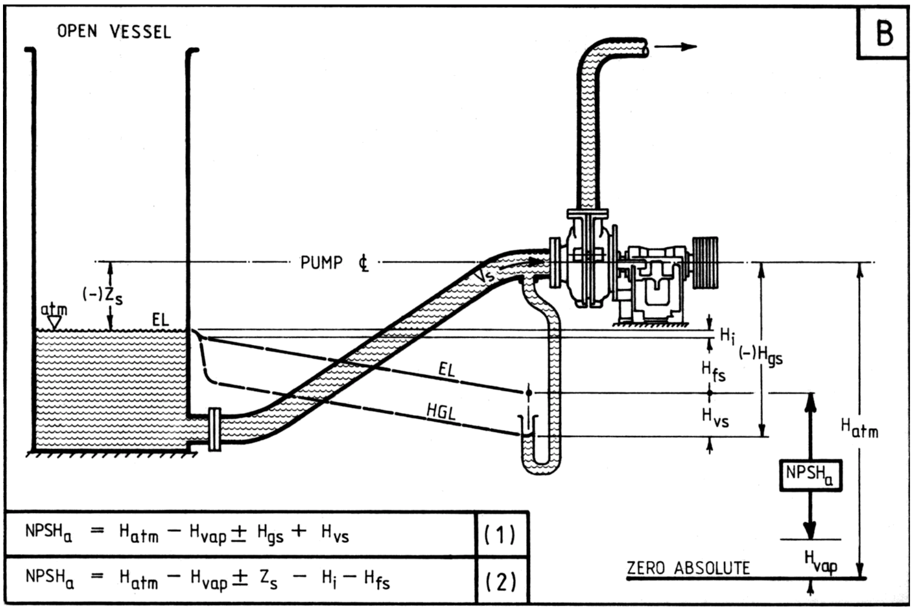


FIGURE A6-1b NPSHa FOR NEGATIVE SUCTION CONDITIONS

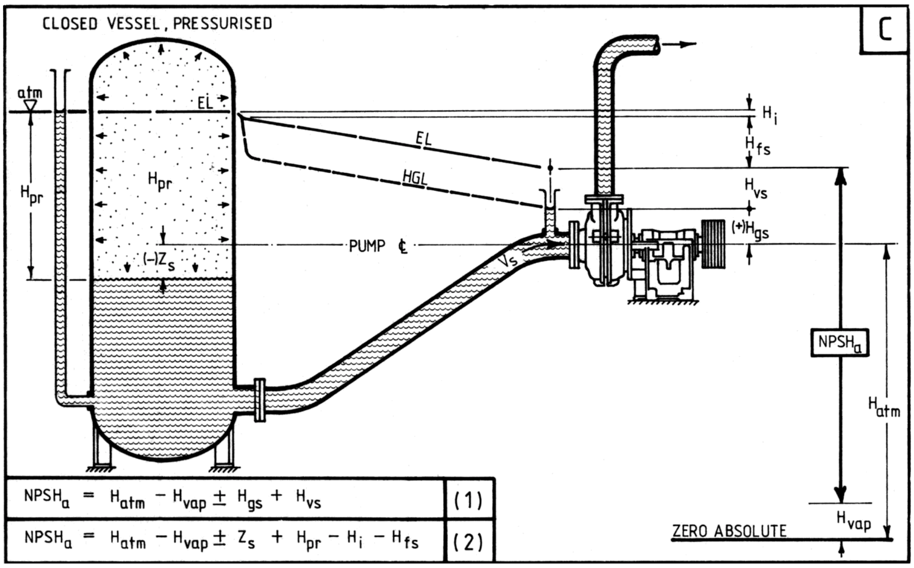


FIGURE A6-1c NPSHa PUMPING FROM A CLOSED PRESSURISED VESSEL

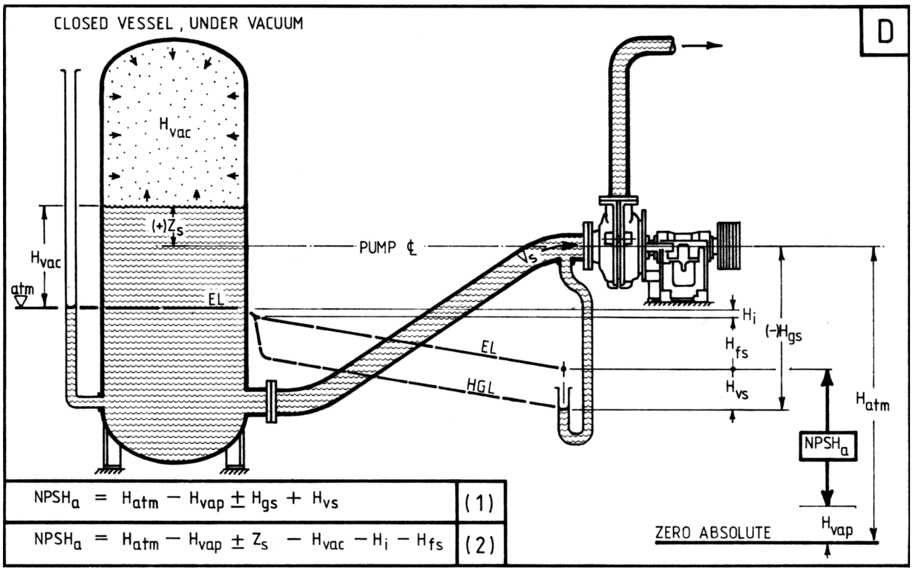


FIGURE A6-1d NPSHa PUMPING FROM A CLOSED VESSEL UNDER VACUUM

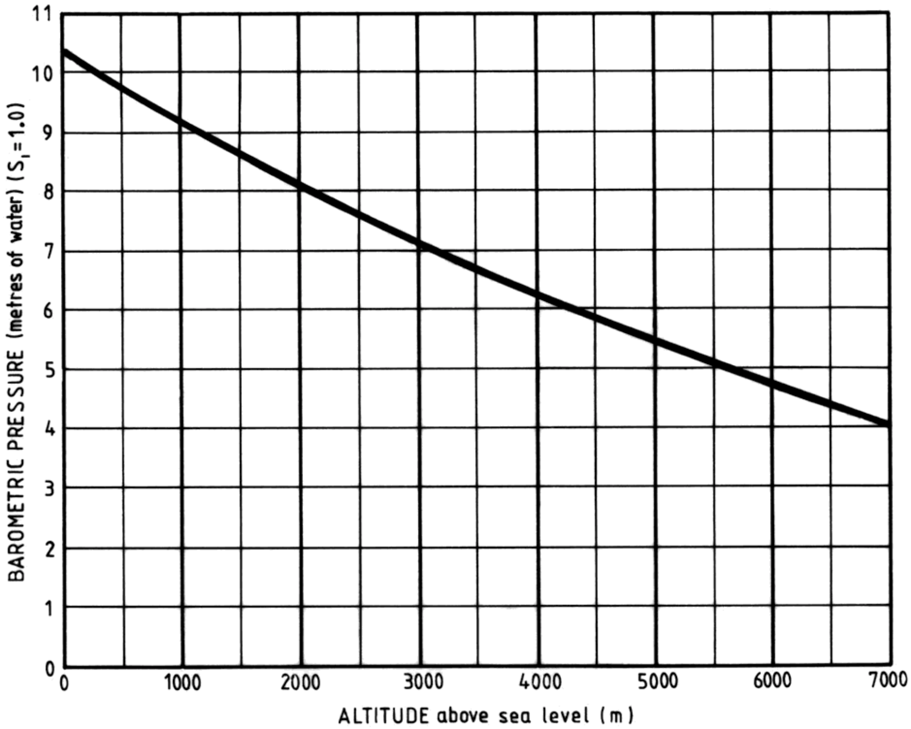


FIGURE A6-2 APPROXIMATE BAROMETRIC PRESSURES

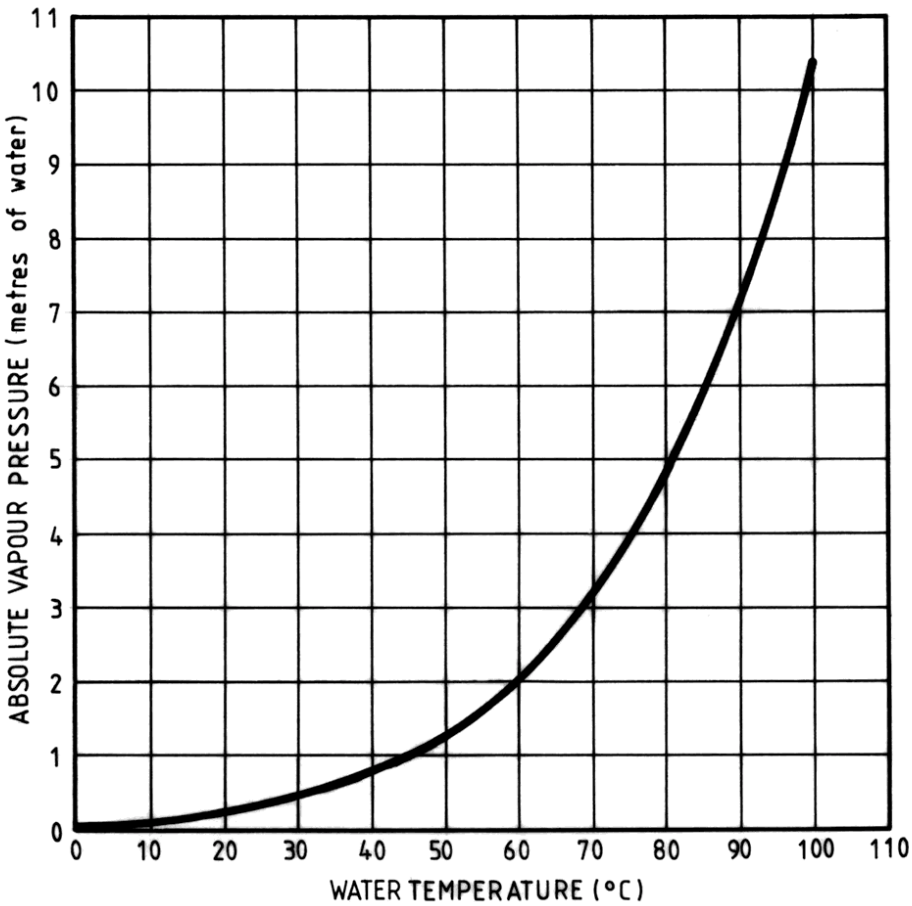


FIGURE A6-3 ABSOLUTE VAPOUR PRESSURE OF PURE WATER

APPENDIX 7 – SERIES PUMPING

GENERAL NOTES

Many pumping duties require slurries to be transported over long distances and/or against very high static discharge heads, for example, against total heads well in excess of heads which can be developed by a single centrifugal slurry pump.

Typical examples include many requirements for the pumping of concentrates, tailings, power station ash and underground fill. The high flow rates required are commonly beyond the capacities of available positive displacement pumps (PDP's). In addition, the overall % efficiency, that is:

$$\frac{(Hydraulic\ (useful)\ power\ imparted\ to\ slurry)}{(Total\ Electrical\ power\ input\ to\ motors)}\ x\ 100\%$$

of large centrifugal slurry pump installations competes with PDP's essentially due to:

a) the high efficiency of large centrifugal slurry pumps, and

b) the higher efficiencies of the lower-ratio drives between electric motors and the centrifugal pumps. These pump applications which require a high Total Head can be handled by series pumping, either as:

   i. multi-stage pump units, or

   ii. separate pumps spaced at intervals along the pipeline route.

SINGLE PUMP

Figure A7-1 represents a single centrifugal pump operating at duty point "A", and at pump speed, n1. For the required flow rate Q1, the pump can develop a head H1 at an efficiency, em1, and at a power consumed of P1. That is, the Duty Point "A" is Q1/H1.

NOTE: The hydraulic grade line (HGL) indicates the actual static head available at any point along the length of the pipeline.

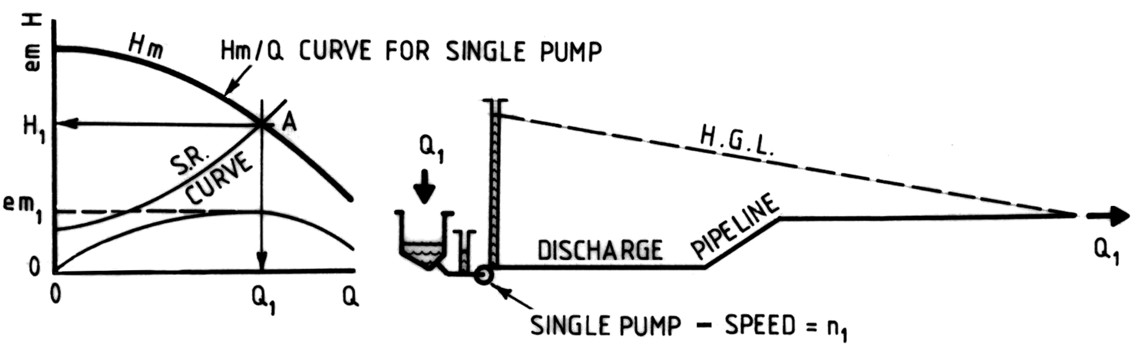


FIGURE A7-1 SINGLE PUMP

Two-Stage Pump Unit

Figure A7-2 represents two identical pumps, arranged in series so that the entire flow discharged from the 1st Stage pump is piped, under pressure through a short length of piping, directly to the suction flange of the 2nd Stage pump and finally from the discharge of the 2nd Stage pump into the discharge pipeline. If both pumps are operated at the same speed, n1 and as both are handling the same required flow rate, Q1, each will develop the same Head, H1, at the same efficiency, em1, and consume the same power, P1.

The total head developed by the 2-stage pump unit combination = 2 x H1, that is,

Duty Point "B" is Q1/2H1

Accordingly, both pumps will be operating under the same conditions, except that the Suction Head and the Discharge Head of the 2nd Stage Pump will both be higher by the value of the Discharge Head of the 1st Stage Pump (less small losses in the inter-stage piping).

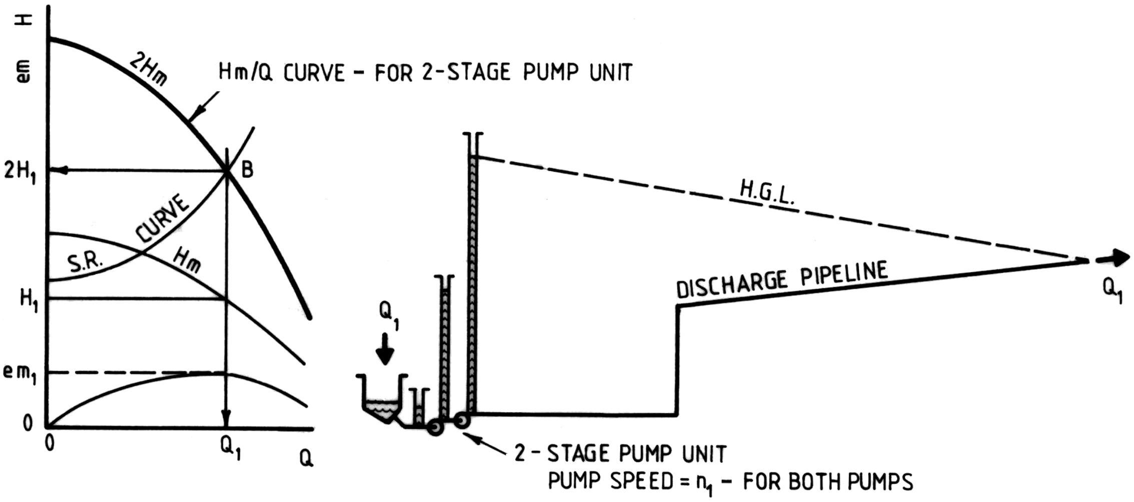


Figure A7-2 Single pump

Four-Stage Pump Unit

Figure A7-3 represents an arrangement similar to Figure A7-2, but extended to represent a 4-Stage Pump unit where the entire flow passes through all 4 identical pumps prior to entering the discharge pipeline.

If all 4 pumps are identical and are operated at the same speed, n1, and as all the pumps are handling the same flow rate, Q1, each will develop the same head, H1, at the same efficiency, em1, and each consume the same power, P1.

The total head developed by the 4-Stage Pump unit combined H1 + H1 + H1 + H1 = 4 x H1, that is, Duty Point "C" is Q1/4HI.

Accordingly, all 4 pumps will be operating under the same conditions except that the Suction Head and the Discharge Head of each successive stage will be progressively higher. Neglecting the small losses in the inter-stage piping and assuming that Hs for the 1st Stage Pump = X (metres) the individual values are:

Stage	Suction Head for Stage (m)	Total Head Developed by Pump (m)	Power Consumed by Each Pump	Discharge Head for Stage = (m)
1st Stage	X	H1	P1	X + H1
2nd Stage	X + H1	H1	P1	X + 2H1
3rd Stage	X + 2H1	H1	P1	X + 3H1
4th Stage	X + 3H1	H1	P1	X = 4H1

4-Stage Unit: Total Head Developed = 4H1;

Total Power Consumed = 4P1

TABLE 2 CALCULATION OF POWER AND HEAD FOR MULTI-STAGE SETS

If the Total Head Developed by each pump varies from one pump to another, due to different speeds or different effects of wear, the Total Head developed by the Multi-Stage Unit will be the <u>sum</u> of the individual total Heads Developed by <u>each</u> of the pumps.

Similarly, the Total Power Consumed will be the sum of the individual powers consumed by each of the pumps.

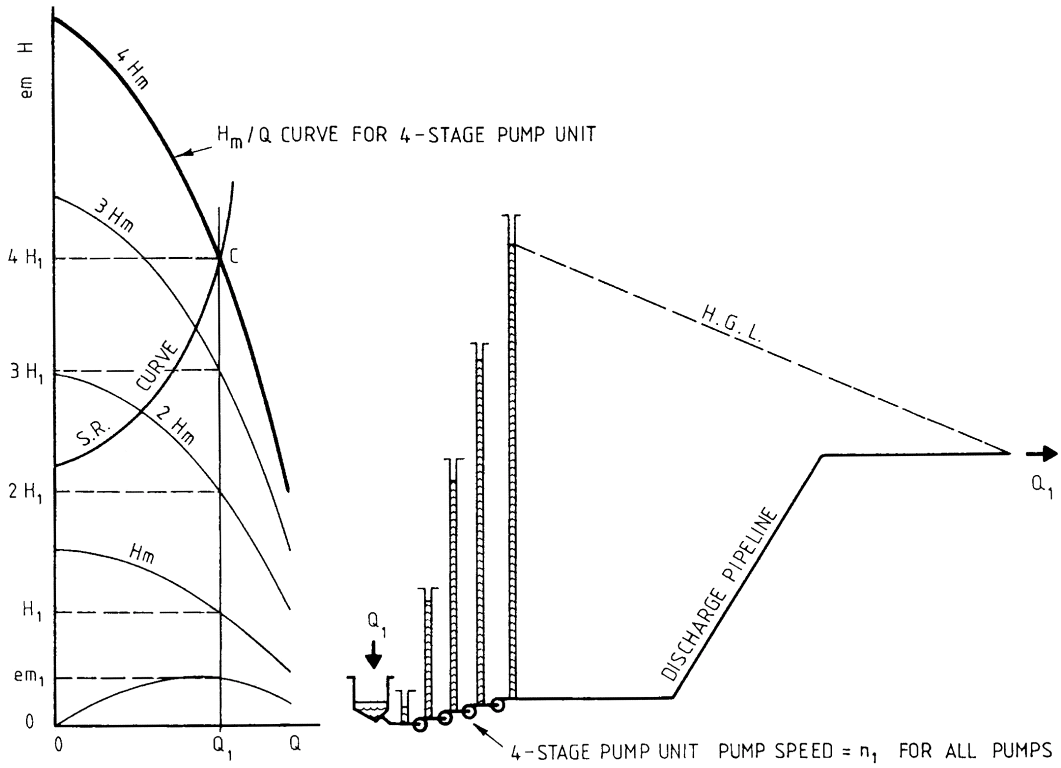


FIGURE A7-3 FOUR-STAGE PUMP UNIT

NOMENCLATURE

Cv	Concentration of solids in mixture, by volume (percent)
Cw	Concentration of solids in mixture, by weight (percent)
D	Inside diameter of pipe (m)
Ds	Inside diameter of suction pipe (m)
Dd	Inside diameter of discharge pipe (m)
d50	Average particle size of solids in a given dry sample. This size is equal to the screen aperture which would retain exactly 50% by weight of the total sample (mm or µm)
em	Efficiency of pump when pumping mixture (percent)
ER	Efficiency Ratio
ew	Efficiency of pump when pumping water (percent)
f	Darcy Friction Factor (dimensionless)
FL	Limiting Settling Velocity Factory (dimensionless)
g	Gravitational constant (9.81 m/s^2)
h	Head symbol utilised for sundry purposes
H	Total dynamic head required by a system: Head of mixture (m)
Hatm	Atmospheric Pressure Head at Pump Location: Expressed as head of mixture pumped (m)
Hd	Total Discharge Head: Head of Mixture (m)
Hf	Friction Head Loss: Head of mixture (m)
Hfd	Friction Head Loss in Discharge Pipe: Head of mixture (m)
Hfs	Friction Head Loss in Suction Pipe: Head of mixture (m)
Hgd	Discharge Gauge Head (above atmospheric pressure): Head of mixture (m)
Hgs	suction Gauge Head: Head of mixture (m)
Hi	Inlet Head Loss: Head of mixture (m)
Hm	Total Dynamic Head Developed by Pump when Pumping Mixture: Head of mixture (m)
HR	Head Ratio
Hpf	Exit Gauge Pressure Head, above atmospheric, at exit from pipeline: Head of mixture (m)
Hpr	Gauge Pressure Head, a above atmospheric pressure, of gas or vapour maintained over mixture surface in a closed supply vessel: Head of mixture (m)
Hs	Total Suction Head: (+ve) or (-ve): Head of mixture (m)
Hvac	Gauge Vacuum Head, below atmospheric pressure, of gas or vapour maintained over mixture surface in a closed supply vessel: Head of mixture (m)
Hvap	Absolute Vapour Pressure head of suspending liquid at pumping temperature: Head of mixture (m)
Hv	Velocity Head, at any given point of evaluation: Head of mixture (m)
Hvd	Velocity Head, in the pump discharge pipe: Head of mixture (m)
Hve	Exit Velocity Head Loss, at final discharge from pipeline: Head of mixture (m)
Hvs	Velocity Head, in the pump suction pipe at suction tapping point: Head of mixture (m)
Hw	Total Dynamic Head, developed by pump when pumping water: Head of mixture (m)
L	Total Equivalent Length of Pipe = La + Lf (m)
La	Total Actual Length of Pipe (m)
Lf	Aggregate of Equivalent Lengths for all Valves, Bends and Fittings contributing to Friction Head Loss in pipeline (m)
Ls	L for Suction Pipe (m) Note: Ls = Las+Lfs
Las	La for Suction Pipe (m)
Lfs	Lf for Suction Pipe (m)
Ld	L for Discharge Pipe (m) Note: Ld=Lad+Lfd
Lad	La for Discharge Pipe (m)
Lfd	Lf for Discharge Pipe (m)
M	Mass flow rate of dry solids (t/h)
n	Pump Rotational Speed (revolutions/minute: r/min or RPM)
NPSHa	Net Positive Suction Head available at Pump Suction Flange: Head of mixture (m)
NPSHr	Net Positive Suction Head required at Pump suction Flange: Head of mixture (m)
NR	Reynolds Number (dimensionless)
P	Power consumed at pump shaft (kW)
Pr	Pressure (Pa)
Q	Mixture flow rate (usually litres per second: L/s)
S	Specific Gravity of Dry Solids
SG	Specific Gravity
Sl	Specific Gravity of Liquid or Transporting Medium
Sm	Specific Gravity of Mixture
V	Average Velocity of Mixture in a pipe (m/s)
Vd	V in Pump Discharge Pipe (m/s)
Ve	V at Exit from pipe (m/s)
VL	Limiting Settling Velocity of mixture (m/s)
Vs	V in Pump Suction Pipe (m/s)
Z	Net Static Head
Zc	Differential Column Head: Head of mixture (m)
Zd	Static Discharge Head
Zl	Vertical height of suction pipe conveying slurry and surrounded by a liquid of Specific Gravity lower than that of the mixture pumped (m)
Zs	Static Suction Head: Vertical height from mixture supply surface level to pump centre-line (m)
Zsm	Effective Positive Static Suction Head above (+ve) pump centre-line: Head of mixture (m)