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Thickened and Paste Tailings Pipeline Systems: Design Procedure – Part 2

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ABSTRACT

This paper is the second in a series examining issues related to the design of paste and thickened tailings pipeline systems. The first paper presented at Paste06 covered design criteria development, characterisation test work and pipeline flow behaviour modelling. This paper discusses centrifugal pump performance derating for non-Newtonian slurries, and hydraulic and mechanical design of paste and thickened tailings systems. The paper concludes by exploring specific aspects of designing thickener underflow pump systems, centrifugal pump systems and gravity backfill systems for Bingham plastic mixtures.

1 INTRODUCTION

The design methodology for pipeline systems conveying thickened and paste tailings systems has been well developed over the last decade. This series of papers outlines the process for designing and implementing a typical surface tailings or underground backfill pipeline system. The papers comprise the following parts:

- Part 1 (presented at Paste06) discussed:
  - development of the design criteria document,
  - issues to be considered for the test work, and
  - pipeline flow behaviour modelling, friction loss calculation and pipe diameter selection.
- This paper covers the following aspects of thickened and paste tailings design:
  - centrifugal pump performance,
  - hydraulic design including hydraulic grade line development, pump suction conditions and transient flow conditions,
  - mechanical design including pipeline thickness, wear and material selection, and pump and motor selection, and
  - considerations for thickener underflow, centrifugal pump and gravity flow systems.

2 CENTRIFUGAL PUMP PERFORMANCE

Centrifugal slurry pumps are able to pump surprisingly viscous thickened tailings mixtures (such as illustrated in Figure 1). The performance of a centrifugal pump is reduced when pumping slurry compared with pumping water. Manufacturers provide clear water pump performance curves which must be derated to account for the effect of slurry when designing a pumping system. Derating parameters are defined for the head developed by the pump and the pump hydraulic efficiency:

$$Head\ ratio = H_R = \frac{H_m}{H_w},\ and \tag{1}$$

$$Efficiency\ ratio = E_R = \frac{\eta_m}{\eta_w}, \tag{2}$$

where H_m = head generated when pumping slurry (metres of slurry)
H_w = head generated when pumping water (metres of water)
η_m = pump efficiency when pumping slurry
η_w = pump efficiency when pumping water.

The above values are determined for a fixed flow rate and pump rotational speed.

Figure 1 **Non-Newtonian thickened tailings (τ_y = 70 Pa, K_b = 0.060 Pa.s)**

Table I presents measured pump performance derating parameters for three typical thickened tailings slurries with varying percentages of coarse particles. The test work has been conducted using a Warman 6/4 pump with a 365 mm diameter impeller. The derating parameters are determined at flow rates corresponding to the pump's best efficiency point (BEP). It is seen that even with yield stresses approaching 100 Pa, the pump performance is not significantly impaired.

Table 1 **Measured pump performance derating (6/4 pump)**

Material / Pump Speed	Slurry Density (kg/m^3)	Yield Stress (Pa)	Plastic Viscosity (Pa.s)	Percentage of particles > 75 µm	H_R	E_R
Slurry 1 1300 RPM	1520 1577	59 95	0.050 0.070	48%	0.96 0.94	0.96 0.96
Slurry 2 1300 RPM	1518 1669[a]	25 70	0.026 0.070	70%	0.95 0.91	0.91 0.91
Slurry 3 1200 RPM	1335	58	0.010	15%	0.98	0.99

Note: [a] This is the mixture depicted in Figure 1.

The standard industry method for derating the performance of pumps handling non-Newtonian slurries is the chart presented in Warman Technical Bulletin Number 14, October 1991 (based on Walker and Goulas, 1994). The head and efficiency derating parameters are considered to be a function of impeller size and rotational speed, and slurry density and plastic viscosity (but not yield stress). These parameters are combined in the form of a pump Reynolds number:

$$Re_p = \frac{\omega \, D_i^2 \; \rho_m}{K_b}, \tag{3}$$

where ω = pump rotational speed (radians/s)
D_i = impeller diameter (m)
ρ_m = mixture density (kg/m^3)
K_b = Bingham fluid consistency index or plastic viscosity[1] (Pa.s).

Figure 2 shows the data presented in Table I plotted as head and efficiency derating versus the Warman pump Reynolds number. The Warman curves slightly under predict the head derating, but significantly over predict the efficiency derating.

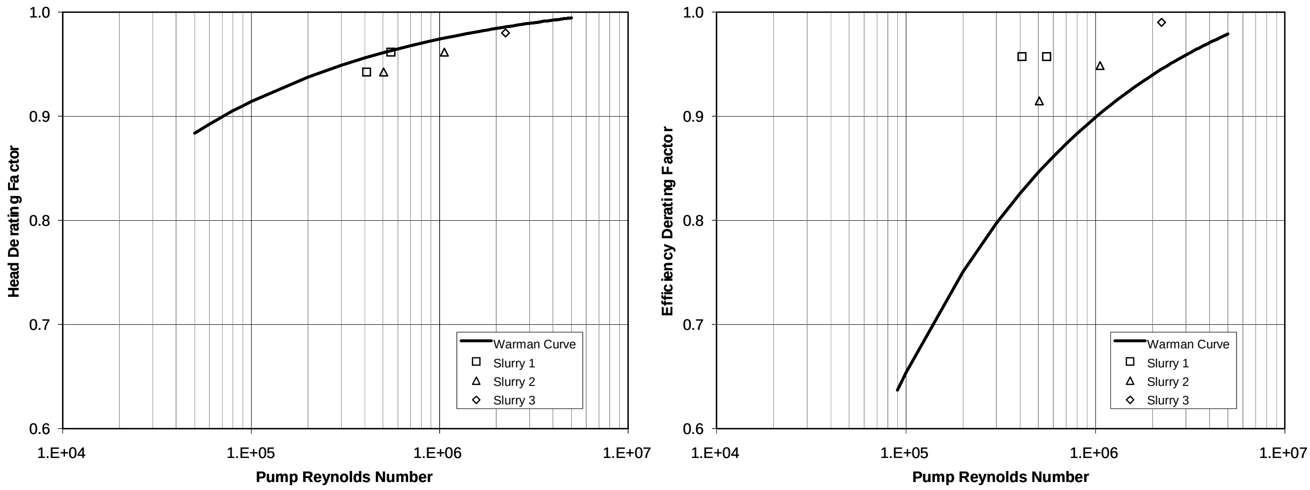

Figure 2 **Non-Newtonian Pump Performance Derating at BEP (Warman Curves interpolated from published curve)**

Selgren et al (2005) and Bootle (2006) report on derating test results for pumps equipped with flow inducing vanes. Selgren's data is for a high yield stress mixture (τ_y = 240 to 350 Pa, K_b= 0.040 Pa.s), while Bootle's results are for intermediate (τ_y = 100 to 200 Pa, K_b= 0.080 to 0.14 Pa.s) and low (τ_y = 1 to 2 Pa, K_b= 0.060 to 0.01 Pa.s) yield stress mixtures. The performance derating measurements are shown in Figure 3.

Selgren's data indicates that the pump performance derating is independent of yield stress for values below about 300 Pa. Above 300 Pa, there is a marked reduction in head generated for flow rates below about 80% of BEP (although there is minimal head reduction at BEP).

The data presented by Bootle shows a surprising trend; the performance derating is greater for the low yield stress mixture than for the intermediate yield stress mixture for a fixed pump Reynolds number (the plastic viscosities of the mixtures are similar).

[1] For low flow rate (10% of the BEP flow rate), Walker and Goulas propose that an apparent viscosity corresponding to a shear rate of 2ω is used instead of the plastic viscosity. This approach is rarely necessary for engineering design use as pumps are selected to operate close to BEP.

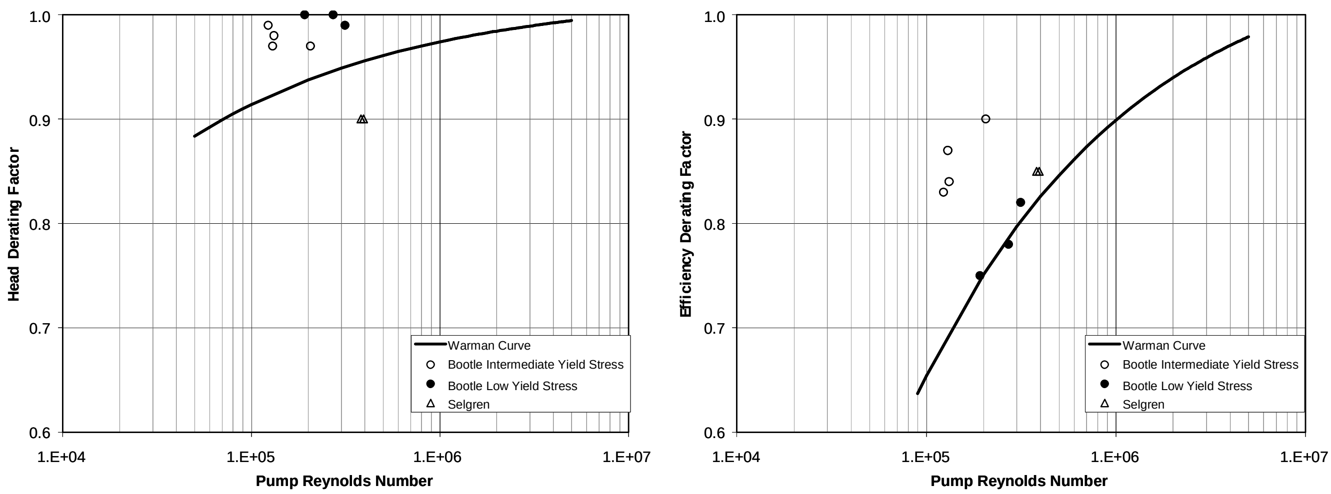

Figure 3 Flow Inducer Impeller Pump Performance Derating at BEP

While further work is required before pump performance derating is properly understood for non-Newtonian slurries, the Warman chart is a useful design tool provided the following points are considered:

- The chart provides a reasonable estimate for head performance derating for Reynolds numbers greater than 3×10^5.
- The chart over predicts the efficiency derating for standard pump designs in sizes greater than the 4/3 pump used to develop the chart. However, it is prudent that the chart derating values are followed for design work due to the likely uncertainty in quantifying the mixture properties. It is expected that the efficiency derating will decrease with increasing pump size.
- Until the derating criteria are better understood, it is recommended that installations are not designed for Reynolds number lower than 3×10^5 without conducting test work. Similarly, the recommended upper limit for yield stress is 200 Pa.
- It is important to operate pumps close to BEP (more so than for other slurry types as operation away from BEP can significantly affect the hydraulic stability of the system).

3 HYDRAULIC DESIGN

The sequence followed for the hydraulic design of a paste or thickened tailings pipeline system is:

- Select the pipe diameter (and trial pipe wall thickness) considering deposition and laminar settling criteria (Cooke, 2006).
- Calculate the unit friction losses.
- Plot the hydraulic grade line for steady state operating conditions.
- Check that the pipe pressure rating meets the steady state pressure envelope, if necessary change the pipe specification and repeat the above steps.
- Determine the hydraulic grade line envelope for transient conditions (pipeline start up, shut down, flushing, etc). This may require a change to the pipe specification and a repeat of the above steps.
- From the hydraulic grade line, determine the pump station duty head envelope.
- Select and specify the pump(s) required for the pump station.
- Check the operating point stability by plotting the pipeline and pump head versus flow rate curves (required for centrifugal pump systems).
- Check that the pump suction pressure is sufficiently high to avoid cavitation.

The discussion below focuses on the development of the hydraulic grade line plot starting with Bernoulli's equation. Pump suction conditions are also discussed.

3.1 Bernoulli's Equation

Bernoulli's equation, or the mechanical energy balance, across two sections in a pipeline system yields:

$$\frac{V_1^{\,2}}{2g} + z_1 + \frac{p_1}{\rho_m\, g} + \Delta H_p = \frac{V_2^{\,2}}{2g} + z_2 + \frac{p_2}{\rho_m\, g} + \Delta h_f \;, \tag{4}$$

where *V* = mean pipeline velocity (m/s)
g = gravitational acceleration (m/s²)
z = elevation (m of slurry)
p = pressure (Pa)
ΔH_p = head input by pump (m of slurry)
ΔH_f = friction losses (m of slurry)
1,2 subscripts refer to the upstream and downstream sections respectively.

Note that the units of each term in the above equation are Joules per Newton; the units are generally referred to as "head" and are expressed in metres of slurry.

3.2 Graphical Depiction of Bernoulli's Equation

Figure 4 presents a graphical depiction of Bernoulli's equation for a pipeline system. The following points are noted:

- The total energy at any section along the pipeline route has three components:
  - potential energy,
  - pressure energy, and
  - kinetic energy.
- These components are all expressed in terms of metres of mixture being transported; elevation head, hydraulic (or pressure) head and velocity head.
- The hydraulic head is the height to which slurry would rise in an open stand pipe as a result of the pressure in the pipeline. The hydraulic grade line is a line drawn through the "hydraulic heads" along a pipeline.
- For a constant diameter pipeline, the energy line (or total energy line) is parallel to the hydraulic grade line and offset by the velocity head.
- The slope of the hydraulic grade line (and the energy line for a constant diameter pipeline) is the hydraulic gradient, i.e. the pipeline friction loss expressed as metres of slurry per metre of pipeline.
- The head generated by the pump is the difference between the suction and discharge piping energy line levels.
- Figure 5 illustrates how a minor loss associated with a fitting is depicted graphically, the loss is expressed in metres of slurry. Minor losses are not significant for typical overland tailings pipelines and underground backfill pipelines, however, they should be considered for in plant piping systems.

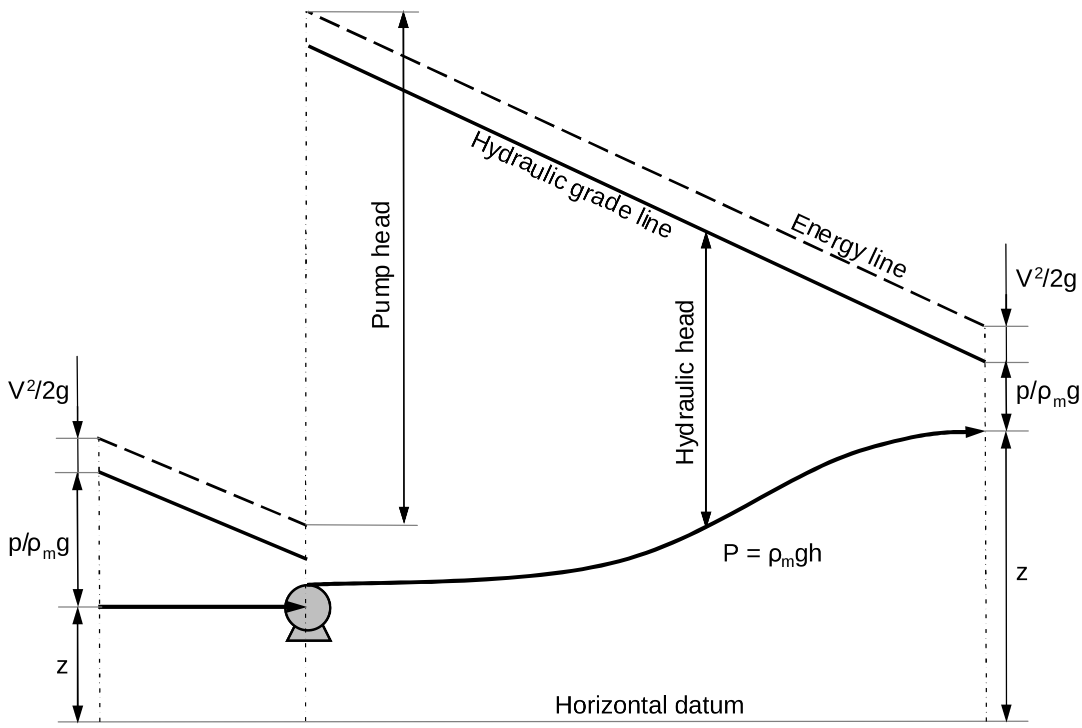

Figure 4 **Graphical depiction of Bernoulli's equation for a pipeline system**

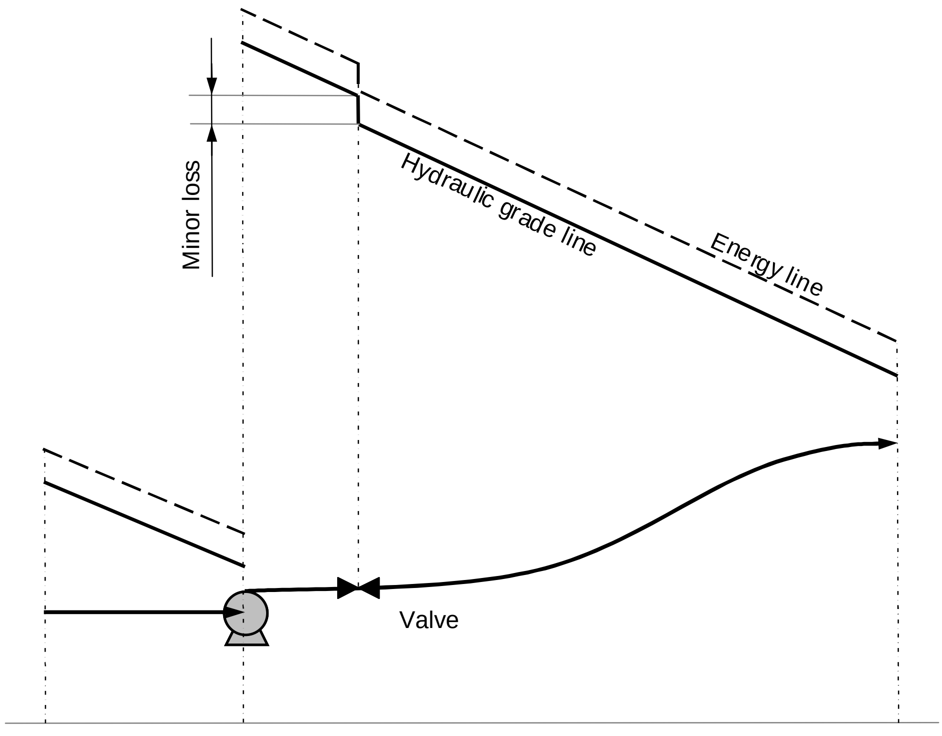

Figure 5 **Depiction of a minor loss due to a fitting (valve)**

3.3 Hydraulic Grade Line Plot

3.3.1 Steady state conditions

As the velocity head is generally small compared with the elevation and hydraulic heads, the kinetic energy component is omitted from graphical engineering depictions of Bernoulli's equation (termed a hydraulic grade line plot as shown in Figure 6). Referring to the plot, the following points are noted:

- The x-axis represents the pipeline length in units of metres or kilometres. The y-axis represents elevation expressed in metres of mixture being transported (the datum elevation is usually taken as mean sea level). Generally the scales of the two axes are different resulting in some vertical exaggeration.
- The pipeline profile is a plot of the actual pipe levels versus pipeline length. Often the ground elevation is also indicated to show where bridges, tunnels and earthworks are required.
- The maximum allowable operating head (MAOH) plot represents the pipeline pressure rating expressed in metres of mixture. The MAOH plot is offset from the pipeline profile by the pipeline pressure rating divided by the mixture density and the gravitational constant.
- Hydraulic grade lines are drawn for the design case, maximum conditions and any other cases that need to be considered in the mechanical design of the pipeline.
- The vertical difference between the hydraulic grade line and the pipeline profile provides a direct indication of the hydraulic head (and thus the pipeline pressure) at any point along the pipeline route. For the case illustrated in Figure 4, it is immediately apparent that the pipeline operating pressure is lower at the pump discharge than it is over the central portion of the pipeline.
- For engineering use, the hydraulic grade line plot will also include the MAOH for transient conditions and the expected operating transient head envelope.

The elevation difference between the hydraulic grade line and pipeline profile is termed the terrain clearance. Care must be taken to ensure that this value remains positive (for long pipelines, the guideline value is between 25 and 100 m). If the hydraulic grade line lies sufficiently below the pipeline profile, the absolute pressure may fall below the slurry's vapour pressure resulting in slack flow.

The hydraulic grade line plot is a valuable design tool, an effective communication technique for depicting the hydraulic design of a pipeline system, and a powerful aid for identifying operational hydraulic problems.

3.3.2 Transient conditions

An envelope of possible hydraulic grade line conditions should be developed for the following cases to identify the most adverse pump station duty and pipeline pressure requirements:

- Start-up and shut down (controlled and emergency) of the pumping system; as high transient pressures may potentially be generated during these events. A computer program is used to perform this simulation and care must be taken that the friction losses are modelled correctly (most commercially available programs do not have the facility to model non-Newtonian pipeline friction losses). The output of this analysis may lead to changes to the system design (route modification, increased pipe wall thickness, incorporation of equipment to minimise transient pressures, etc).
- Start-up of the system with a full (water and slurry) and empty pipeline.
- Pipeline flushing.

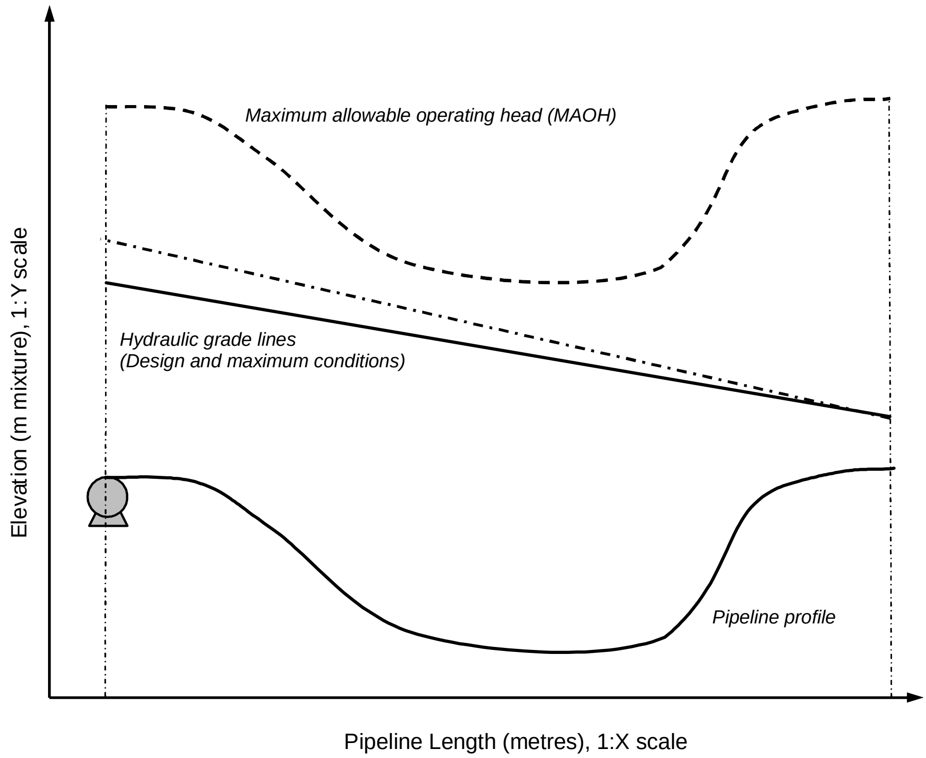

Figure 6 Hydraulic grade line plot

3.4 Pump Suction Conditions

3.4.1 Centrifugal pumps

For conventional slurry pumping systems, the net positive suction head (NPSH[2]) available must be greater than the NPSH required to avoid cavitation. Pumps can operate reliably with negative gauge suction pressure (e.g. dredging applications). However, for thickened and paste tailings this issue is more complex:

- Due to the typically high viscosities, there is a high likelihood of air being entrained into the pump suction (particularly if the discharge into the sump is above the mixture level). The presence of air in a centrifugal pump causes a substantial reduction in the head generated by the pump. Note that this phenomenon is not cavitation, but rather a head derating effect due to the lower mixture density.
- Laboratory tests indicate that the vapour pressure for viscous high concentration water based mixtures is equal to the value for water alone[3].
- Bootle (2006) notes that for Bingham Plastic mixtures, due to the modified velocity distribution in the pump, the NPSH required by the pump can be significantly greater than the NPSH required when pumping water. This effect is likely to be more marked for smaller pumps.

The following guidelines should be considered:

- Minimise the possibility of air entrainment into the mixture through careful sump design.
- Minimise the suction piping friction losses.
- Avoid operating with negative gauge pressures, it is suggested that a positive head of at least 2 m at the pump inlet is provided.

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[2] NPSH is defined as the total absolute head at the pump centre line less the mixture vapour pressure head expressed in metres of slurry.

[3] L Francis, unpublished research.

- Avoid using pumps smaller than 4”/3” for mixtures with yield stresses exceeding 200 Pa. The efficiency derating may be significantly higher than indicated by the Warman non-Newtonian slurry derating chart. Peristaltic pumps may be a better option for these duties.

3.4.2 Positive displacement pumps

Positive displacement pumps require a positive gauge pressure on the suction side on the pumps. This is either provided through an elevated feed sump (concrete piston type pumps) or though the use of a charge pump (piston pumps, with and without a diaphragm). This pressure is dependant on the mixture properties and the pump configuration and it is recommended that advice is obtained from the pump vendor.

4 MECHANICAL DESIGN

4.1 Pipelines

The mechanical design of the pipeline involves selecting the piping materials, wall thickness, and the support system. Slurry pipelines are typically designed in accordance with ANSI/ASME Code B31.11.

4.1.1 Pipe materials

The primary piping materials used for paste and thickened tailings pipelines are carbon steel and high density polyethylene.

Steel piping is suitable for all practical operating pressures and a wide range of diameters, wall thicknesses, steel grades, flanges and fittings are available. A variety of internal lining materials can be used to maximise the pipeline life, e.g. rubber, polyurethane, polyethylene and even mortar. Often the pipeline is protected against external corrosion through the use of a coating or wrapping.

High density polyethylene piping is ideal for low pressure applications. Care must be taken to derate the pipeline pressure rating due to effects of wear and temperature.

4.1.2 Support system

Paste and thickened tailings pipelines are generally constructed above ground. A support system comprising supports, guides and anchors is required to cater for pipe movement and loads imposed on the pipeline as shown in Figures 7 and 8. For high pressure systems, a stress analysis is undertaken to demonstrate code compliance.

Figure 7 Typical surface tailings pipelines (expansion loop to cater for temperature variation)

Figure 8 **Typical backfill pipe supports (top: anchors, bottom supports; left: surface, right underground)**

4.2 Pump Stations

The mechanical design of a pump station involves specifying all equipment and piping in the station. Careful consideration must be given to the layout of a pump station to ensure that suitable access and lifting equipment is provided for maintenance.

4.2.1 Centrifugal pumps

Centrifugal pumps may be used in series for discharge pressures up to about 4 MPa. The following points should be considered when designing a centrifugal pump station:

- The pump train configuration is selected to optimise the station footprint while maintaining good accessibility; parallel and right angle shaft configurations can be employed.
- The provision of variable speed pumps, and if so, on which stage the variable speed drive is fitted. The cost of variable speed drives is reducing, so it is likely that in the future all pumps in a train will be equipped with variable speed drives.
- The type of gland sealing arrangement provided; water flushed or mechanical seal. Mechanical seals are ideal as there is no dilution of the tailings slurry, the reliability of these seals is improving and the cost is reducing and so their application is likely to become more widespread.

4.2.2 Positive displacement pumps

Positive displacement pump stations are typically specified for discharge pressures in the range of 4 to 25 MPa. As the pumps have a maximum volumetric capacity of about 800 m^3/h, multiple pumps are installed in parallel for high flow rate applications.

The pump station layout is to a large extent governed by the need to ensure that the high pressure piping has sufficient flexibility.

Safety considerations specific to positive displacement pump stations are:

- Rupture or pressure relief devices are installed to protect the pump against fault conditions. Care must be taken to ensure, that in the event of an over pressure, the discharge is directed to a safe location.
- Care must be taken to ensure that pressure energy which may be trapped in the pulsation dampeners is safely released before any maintenance on the pump.

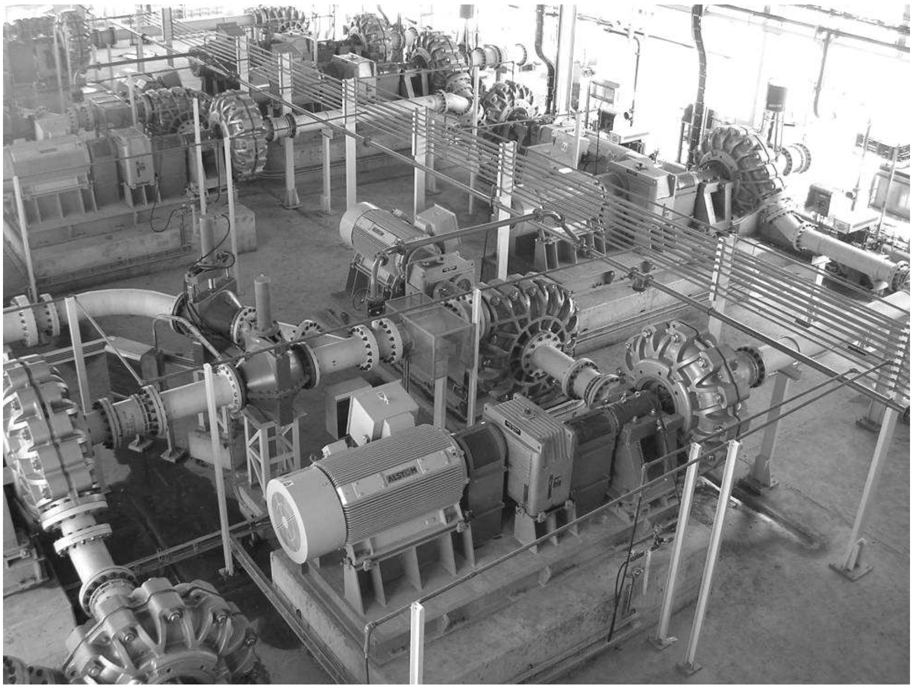

Figure 9 Typical centrifugal pump station

5 SYSTEM SPECIFIC CONSIDERATIONS

5.1 Thickener Underflow Systems

It is often difficult to accurately control the discharge from a thickener using the underflow pump for Bingham plastic slurries. Figure 10 illustrates an example of an underflow pumping system where if conventional pump selection criteria are followed an 8/6 pump will be selected for the duty. Both the pipeline system curve and the pump curve are relatively insensitive to flow rate resulting in a shallow intersection angle between the pump and system curves; a small change in the slurry properties or the pump rotational speed, will result in a large change in the underflow flow rate. Selecting a smaller pump (operating just to the right of BEP) results in a more stable operating point as illustrated for a 6/4 pump.

An alternate approach is to specify smaller diameter piping operating at high velocities. This approach is beneficial if it results in turbulent pipeline flow (the operating point will be stable as the friction losses will be relatively insensitive to changes in the slurry properties). If the flow remains laminar, there is little impact on the operating point stability.

For low flow rate systems, accurate control can be achieved using peristaltic pumps.

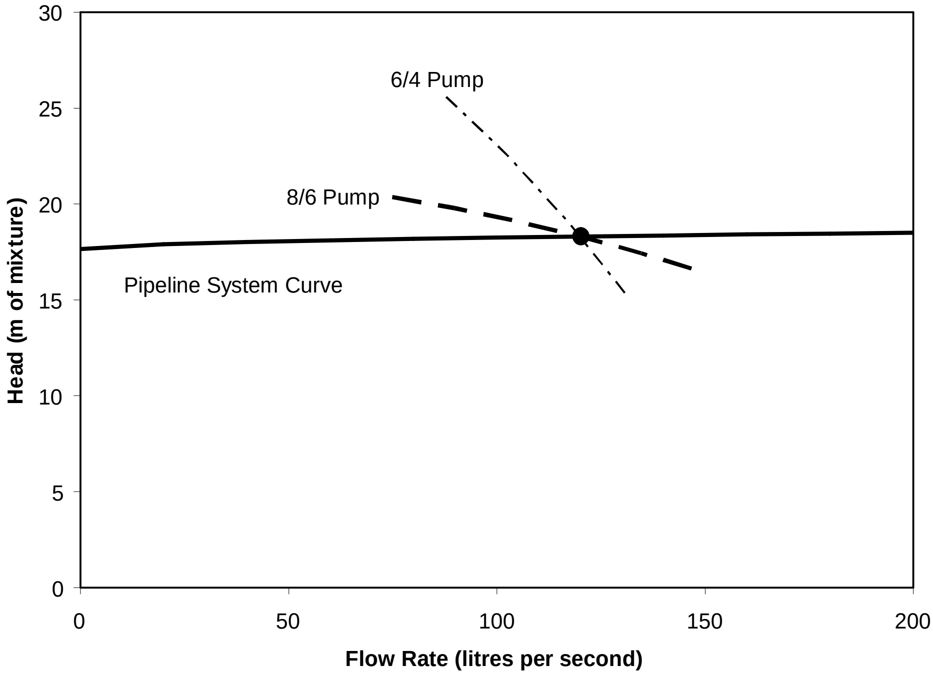

Figure 10 Thickener Underflow System Curves

5.2 Centrifugal Pump Systems

As centrifugal pumps generate head and not pressure, there can be a substantial reduction in pump station discharge pressure when flushing is initiated (related to the change in density less the head derating effect). For high yield stress Bingham plastic slurries operating in laminar flow, it is possible that the pumping system will stall. This can be resolved by increasing the speed of the pumps and/or introducing high pressure flush water on the suction side of the pump train.

5.3 Underground Paste Backfill Systems

The hydraulic design of underground paste backfill systems is challenging:

- Paste backfill systems operate in laminar flow resulting in a flat system curve (i.e. the friction losses are insensitive to flow rate). The pipeline friction losses are often highly sensitive to small changes in paste properties. These factors coupled with the flat characteristic of the gravity head driving the flow, results in a potentially unstable operating point.
- Many systems use a piston type pump to introduce paste into the pipeline system and provide operating point stability. Care must be taken when specifying the maximum pump discharge pressure; a small change in the pipeline friction losses may result in a substantial change in the required pump delivery pressure.
- The pipeline routing is largely dictated by the mining requirements. This results in the designer having to deal with unfavourable pipeline profiles which may require the hydraulic grade line to be manipulated through the use of choke or energy dissipation stations.
- The start up of deep mine gravity flow systems is potentially problematic due to the high velocities that can arise when starting to fill an empty pipeline. Care must be taken to carefully specify the system start up and shut down procedures.

ACKNOWLEDGEMENTS

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