A De-Rating Method for Centrifugal Pumps Pumping "Paste"

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ABSTRACT

Centrifugal pumps are widely used for transporting suspensions but their performance is de-rated when non-Newtonian fluids, such as pastes, and/or coarse solids are present. This paper presents a modification of the Hydraulic Institute de-rating method suitable for any homogeneous non-Newtonian rheology The method based on paste rheology, pump geometry and flow rate through the pump predicts the dramatic characteristic reduction in head at low flow rates that is often observed and explains why larger pumps are relatively insensitive to this form of head de-rating. Where the pastes contain large particles, e.g. in co-disposal systems, it is shown that the de-rating due to the presence of the coarse particles often dominates that due to the viscous nature of the paste. Data is presented for a wide range of paste rheologies and solids concentrations.

1 INTRODUCTION

While paste and thickener underflow pumping is often associated with positive displacement pumps centrifugal pumps are also widely used. Pastes are often erroneously thought to behave as homogenous suspensions and while the fine particle pastes (d < 20 μm) may be considered to act like this, the wider size distribution suspensions instead behave as a complex suspensions comprising a non-Newtonian carrier fluid and non-interacting coarse solids. Plant designers usually only have the water performance of the pump available for selection and an effective generalised method to allow calculation of pump de-rating for non–Newtonian fluids and complex suspensions is required.

For Newtonian fluids, the effect of viscosity on centrifugal pump's performance has been well established and methods like the Hydraulic Institute method (1969) are used everyday. Non-Newtonian de-rating however, is more complex, as the fluid's viscosity is dependent upon the local shear rate, and determining a suitable representative value for geometry as complex as a centrifugal pump is a very difficult task.

Similarly, for Newtonian suspensions the effect of coarse particles on centrifugal pumps' performance has been the subject of many studies, e.g. see Engin and Gur (2003) and there are only a very few researchers (e.g. Sellgren et al., 1999) who consider solids conveyed in a non-Newtonian carrier.

The range of combinations of non-Newtonian behaviour, solids and pump geometry unfortunately prevents a generalised method from being presented here, and only two pump series are considered, but the analysis to follow does provide some insight into the mechanism of pump de-rating and provides a basis for more pumps series to be included.

Pastes are typically thought of as having yield stresses of the order of 100s of Pa. However, such yield stresses are often derived from slump or vane tests on un-sheared materials. These yield stresses, often known as static yield stresses, are important in determining deposit integrity but usually exceed the dynamic yield stress that is required to describe the paste's flow behaviour in a pipe and do not take into account any thixotropy in the material Furthermore the presence of coarse particles, described above, will erroneously increase the perceived yield stress. In the study presented here the "paste" is separated into an underlying carrier fluid and tests are conducted with this alone, and also for the combined effect of the coarse particles and carrier fluid. Whilst the yield stresses of the fluids considered in the present work are not in the 100s of

Pa range, it is expected that this approach should also be suitable for higher yield stress materials and should be tested in due course for such materials.

2 TEST FACILITY

Two centrifugal pumps, a Warman 4x3 AH and a GIW 4x3 LCC-M80-300, were tested in the CSIRO's pipeline facility's pump station which is shown schematically in Figure 1. The main dimensional details of each pump are shown in Table 1. The differential pressure transducer across the pump was a Rosemount 3051 CD with a range of 300 kPa. Flow rate was measured with a Fisher and Porter Minimag (IOD 1475) magnetic flow meter fitted to the outlet flow line.

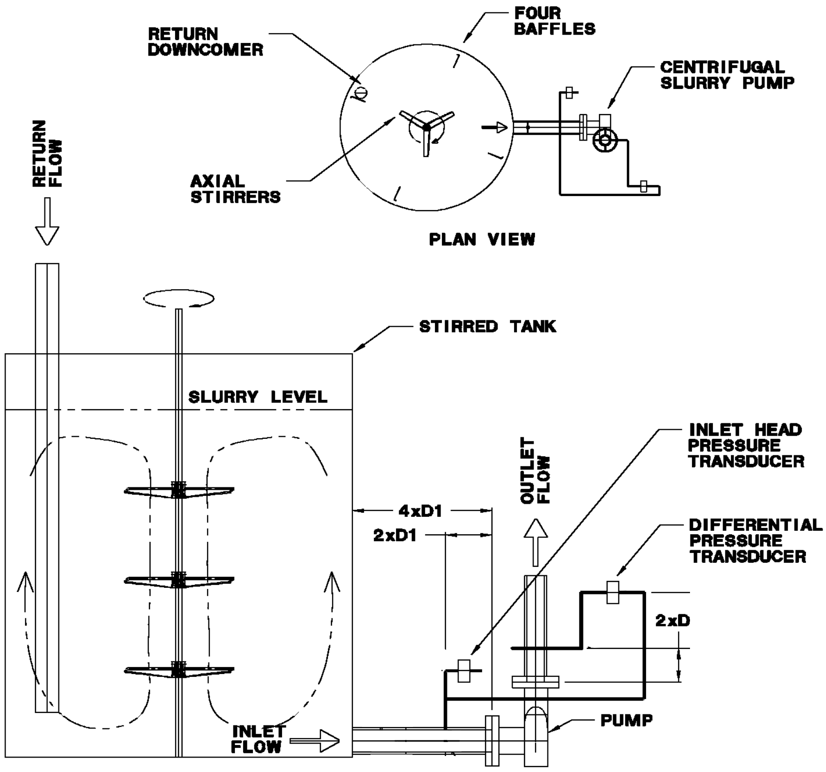

Figure 1 Test facility

Table 1 Pump dimensions

Pump	Impeller diameter D_{imp} (m)	Number of vanes N_B	Characteristic dimension w (m)	Ratio w/D_{imp}
Warman 4x3 AH	0.245	5	0.035	0.143
GIW 4x3 LCC-M80-300	0.310	3	0.045	0.147

The non-Newtonian fluids used were aqueous polymer solutions: CMC (approximating a power law fluid) and Ultrez 10 (approximating a Herschel-Bulkley fluid). These fluids were chosen as their rheologies are typical of many thickener underflows and carrier fluids found in industry. Complex suspensions were made up by adding crushed glass or sand to these carrier fluids. The characteristics of the suspensions were determined by measuring the carrier fluid rheology and particle size distribution by screening, where applicable. A Bohlin CVO 50 controlled stress viscometer was used to determine the rheological properties of the CMC and Ultrez carrier fluids.

3 PUMP DE-RATING

The effects of viscous and coarse particle pump de-rating have been separated in the analysis presented here by using coarse particles that are too large to influence the properties of the underlying carrier fluid. This approach has successfully proved to model more continuous wide size distributions found in the industry (Pullum et al., 2006).

3.1 Modified Hydraulic Institute Method

Viscous pump de-rating, typified by the Hydraulic Institute method, requires a characteristic viscosity for its application. For Newtonian fluids this is simply the dynamic viscosity, but for non-Newtonian fluids, typical of thickener underflows and fine pastes, the viscosity is a variable function of the local fluid shear or strain rate. Generally this will vary throughout the pump and with operation, and finding a representative viscosity that is applicable to the entire range of operation, similar to a Newtonian viscosity, is the goal of many approaches. In an attempt to do this for Bingham plastic fluids, Walker and Goulas (1984) suggested that for high pump Reynolds number flows the plastic viscosity of the fluid should be used as the representative value. While the Bingham plastic model has no theoretical basis, this approach is quite valid for high Reynolds number, or high shear rate flows, as the plastic viscosity asymptotically approaches the actual high shear viscosity and so is a good approximation. At lower Reynolds numbers they suggested that a viscosity based on a characteristic shear rate equal to twice the angular frequency of the pump be used. This was in recognition of the fact that at lower flow rates, often characterised by lower pump speeds, the viscous de-rating was greater implying a higher viscosity. Sery and Slatter (2002) also followed this approach and were able to correlate the data in a similar manner to Walker and Goulas. Sery et al. (2006) suggested that the viscosity of the fluid in the pump could be evaluated using a technique developed by Metzner and Otto (1957) for mixers in standard mixing tanks. This method also suggests that the shear rate is a simple function of the impeller's rotational speed. In a standard mixing tank the flow through the agitator's blades is a function of impeller geometry, rotational speed and fluid properties alone. Unfortunately, in a centrifugal pump, by contrast, the flow through the impeller is also a function of the pipeline's system curve, e.g. shutting a discharge valve downstream of a pump will reduce the flow through the pump, and hence the fluid shear rate, even if the impeller speed is constant.

In this paper we suggest that a more consistent approach is to calculate the viscosity based on a local shear rate, and use a hydraulically equivalent pipeline as a model of the impeller and pump passages to obtain this viscosity. To simplify the analysis, the equivalent hydraulic diameter of this pipe is based on the impeller's dimensions and a characteristic dimension, w, obtained from a global non-linear fitting procedure for a particular fluid and pump. These values are then used for other materials and other geometrically similar pumps. The procedure is as follows:

The equivalent "pipe" diameter, D_h, is calculated from:

$$D_h = \frac{4w\pi D_{imp}}{2(\pi D_{imp} + w)} \tag{1}$$

where w is a characteristic dimension to be determined experimentally as described in §3.2. Note this ignores the impeller vane thickness, but since a characteristic geometry is sought such refinement is not justified.

The velocity, V, through the hydraulically equivalent pipe, D_h is found from:

$$V = \frac{4Q}{\pi D_h^2} \tag{2}$$

where Q is the bulk flow rate through the system.

If the flow is laminar, the shear rate is then obtained from the Rabinowitsch-Mooney relationship:

$$\dot{\gamma} = \left(\frac{3n'+1}{4n'}\right)\frac{8V}{D_h} \tag{3}$$

where n' is the gradient of the curve dln(τ_0)/dln($8V/D_h$) obtained from the rheological model (or rheograms) of choice and to is the wall shear stress calculated at the pseudo shear rate $8V/D_h$.

If the flow is turbulent an asymptotic high shear rate viscosity η_∞ is used – at a shear rate typically in excess of 4000 s^{-1}.

The resulting viscosity $\eta(\gamma)$ or η_∞ is then used with the Hydraulic Institute de-rating method as usual.

Note this technique also uses a high shear viscosity for turbulent flows, but since turbulent flows are very weak functions of viscosity, viscous de-rating under these conditions would be expected to be low or non-existent.

Based on the dimensions of the rotor passages, the expected flow regimes in the pump series tested here are shown in Figure 2 for moderate to low viscosity slurries.

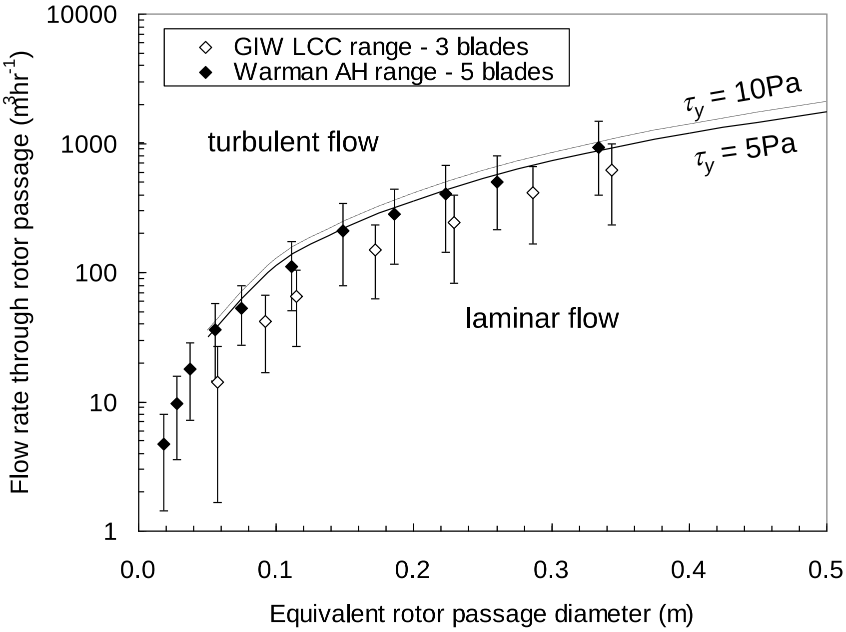

Figure 2 **Flow regimes for low to moderate pastes in two common pump series**

Each pump from the series, e.g. 6 x 4, 8 x 6 etc. is shown as a point with vertical lines extending to the normal flow rate range for each pump. As seen, even for these relatively low viscosity pastes, the flow in the impeller passages, and hence the rest of the pump can be expected to be laminar for most, if not all of the pumps' operating ranges. Consequently viscous de-rating is expected to occur.

3.2 Viscous De-rating: Results and Analysis

Determination of the characteristic dimension, *w*, for a centrifugal pump, currently requires experimental pump head data for a non-Newtonian fluid of the expected range of rheologies. The correction procedure detailed above is applied to the experimental data and a global non-linear minimisation procedure used to determine a value of the characteristic dimension, *w*, which minimises the error between the actual non-Newtonian fluid data, and that calculated with the de-rating method for these data sets. This characteristic dimension can then be used for other non-Newtonian fluids being pumped by the same pump.

The details of the fluids' rheologies for the tests reported here are given in Table 2.

Table 2 Details of the pump tests and fluids for determination of characteristic dimension

Pump	Fluid	τ_y	*k*	*n*
GIW 4x3	CMC	0	5.8	0.48
GIW 4x3	Ultrez 10	14.0	11.0	0.39
GIW 4x3	Ultrez 10	17.0	7.7	0.43
Warman 4x3	CMC	0	8.5	0.46
Warman 4x3	CMC	0	4.5	0.48
Warman 4x3	CMC	0	8.6	0.45
Warman 4x3	CMC	0	4.0	0.47
Warman 4x3	CMC	0	13.9	0.41
Warman 4x3	Ultrez 10	12.2	10.8	0.39
Warman 4x3	Ultrez 10	17.2	18.5	0.35

Results from the two pumps tested are shown in Figure 3 where the predicted head at the same flow rate is shown plotted against the actual head for the various non–Newtonian fluids.

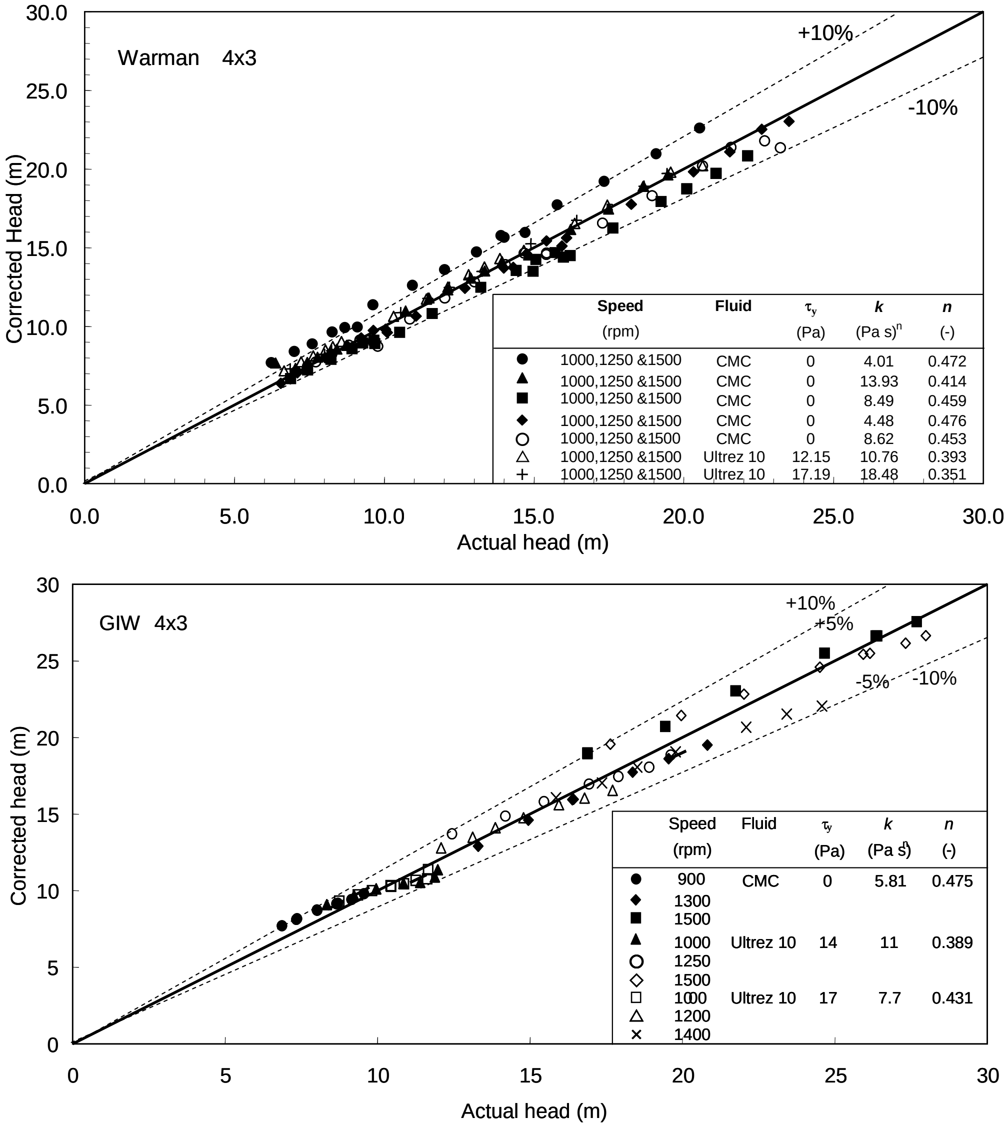

Figure 3 Corrected head versus the actual head

Typical results of the predicted head versus flow rate once the characteristic dimension, *w*, has been globally determined are shown in Figure 4.

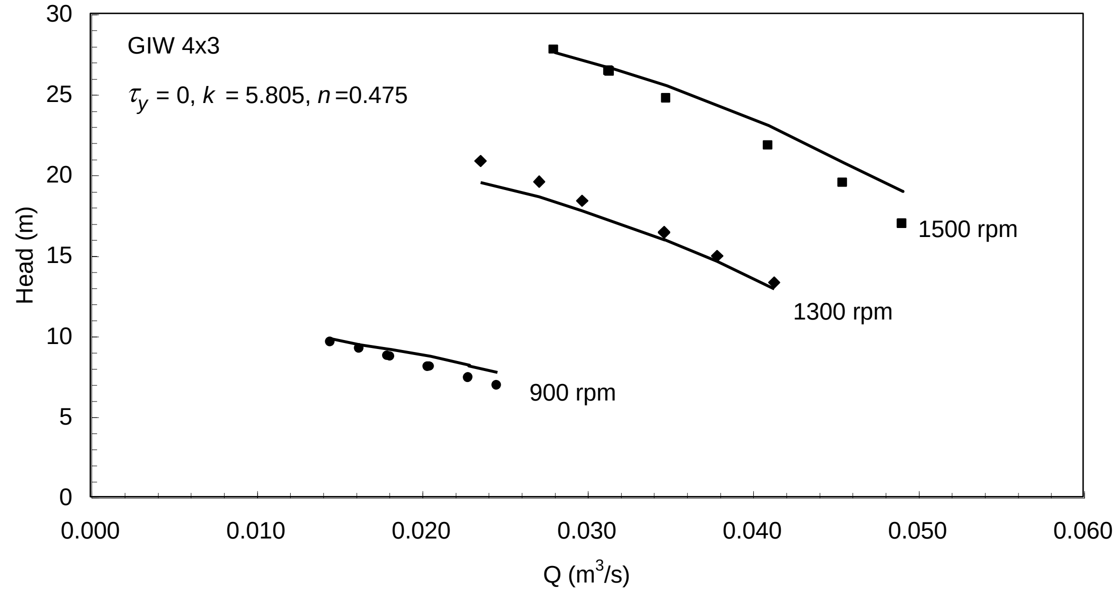

Figure 4 Head flow curves for the GIW 4x3 pump and CMC fluid. Predictions are shown as solid lines

Using this method over the entire flow rate range of the pump produces the characteristic droop in the pump's performance at low flow rates as shown in Figure 5.

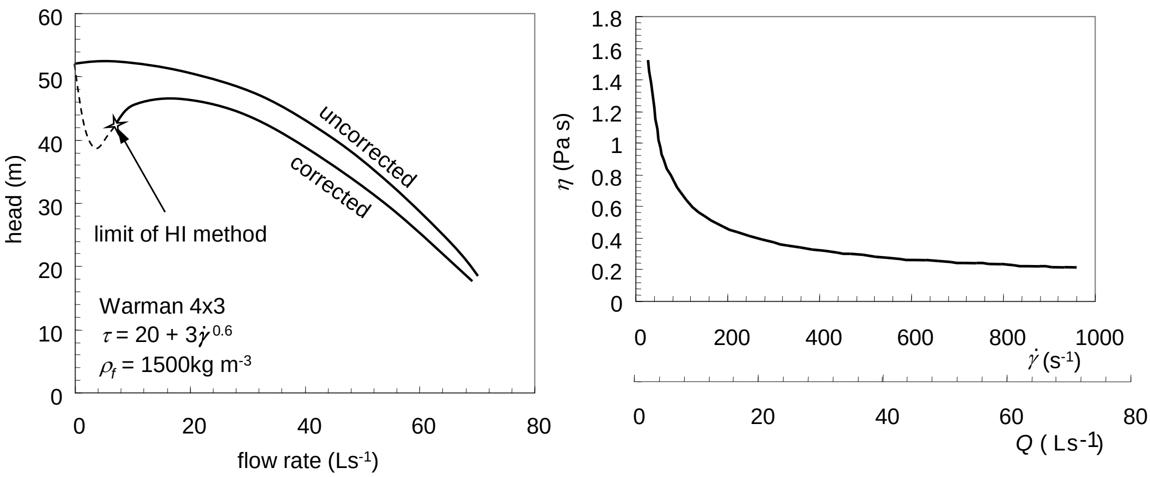

Figure 5 Predicted head flow curve for the Warman 4x3 for a yield visco-plastic fluid and the corresponding fluid rheograms

The extremely shear thinning nature of the fluid is shown in the rheogram displayed in the right hand pane of the figure. Beneath this rheogram are estimates of the corresponding flow through the pump. As the flow rate reduces, the viscosity of the fluid increases, and hence the head reduction calculated by the Hydraulic Institute method increases. For this particular combination of pump geometry and fluid rheology the valid extent of the Hydraulic Institute method extends to flow rates down to around 8 Ls^{-1}. Beyond that, the de-rating still increases as indicated by the dashed line. At pump shut off, however, the pump will still deliver the full shut of head and so the curve must ultimately rise again as shown.

Similar de-rating is expected to occur for larger pumps, but depending upon the geometry and fluid rheology, the flow may become turbulent, since the Reynolds number is directly proportional to pump size. Under these conditions the flow is only a very weak function of the viscosity of the fluid and so head de-rating tends to zero.

Because this method requires the establishment of a characteristic dimension its general applicability to other pumps is yet to be proven. The present results have shown that the characteristic dimension was a similar fraction of the impeller diameter for both pumps considered here as shown Table 1. Thus in principle it should be possible to extend this method to other pumps if it is validated by tests of further pump types with non-Newtonian fluids.

Data obtained at Cape Peninsula University of Technology, South Africa, for kaolin suspensions pumped with a similar GIW 4x3 pump are shown below where the agreement is seen to be similar to that obtained in the CSIRO laboratories.

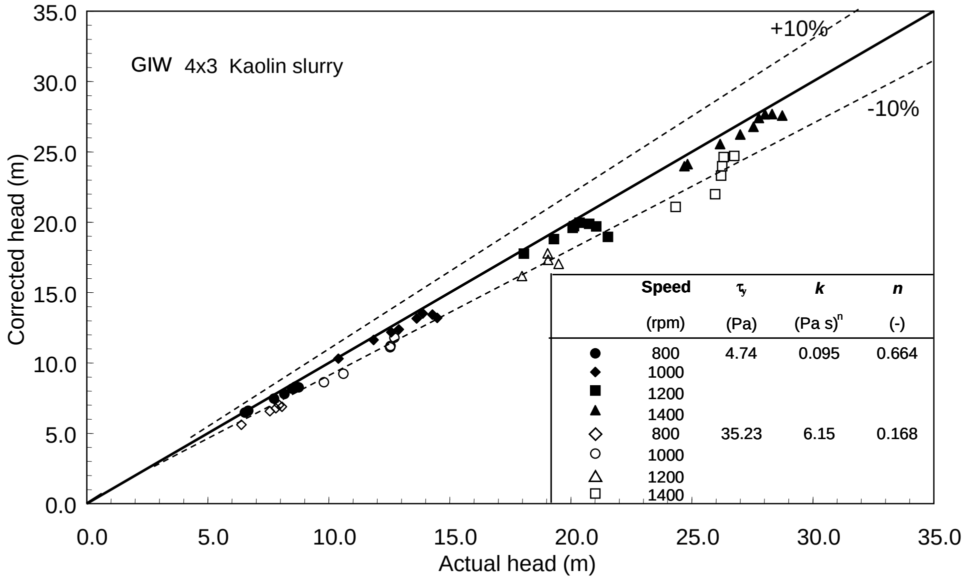

Figure 6 Corrected head versus actual head for a GIW pump pumping kaolin suspensions

3.3 Coarse Solid De-rating

Pump performance experiments were carried out to determine the effect of adding coarse solids to non-Newtonian carrier fluids. The suspensions, pumped by the Warman 4x3 pump, are detailed in Table 3 and used either sand or crushed glass with sizes ranging from 1.1 mm to 7 mm.

The total head de-rating for the suspensions was from 10% – 40% for all suspensions tested and this included the carrier fluid's viscous de-rating as well, which ranged from 5 to 25%.

Typical results for power law fluids are shown in Figure 7 where the head de-rating is seen to be a strong function of the solids concentration and follows similar behaviour to that found with water-based suspensions. Similar behaviour was also observed for the visco-plastic fluids.

Table 3 Parameters for solid de-rating tests

				Carrier Fluid Properties		
Fluid	Solid	d_{50} (mm)	c_v (-)	τ_y (Pa)	k(Pa.s^n)	n(-)
CMC	Sand	1.56 – 2.24	0.05 – 0.3	–	0.25 – 5.33	0.531 – 0.789
CMC	Glass	1.12 – 1.84	0.1 – 0.3	–	1.87 – 3.51	0.494 – 0.552
Ultrez 10	Glass	1.41 – 3.33	0.1 – 0.3	3.6 – 14.6	1.10 – 11.05	0.369 – 0.615

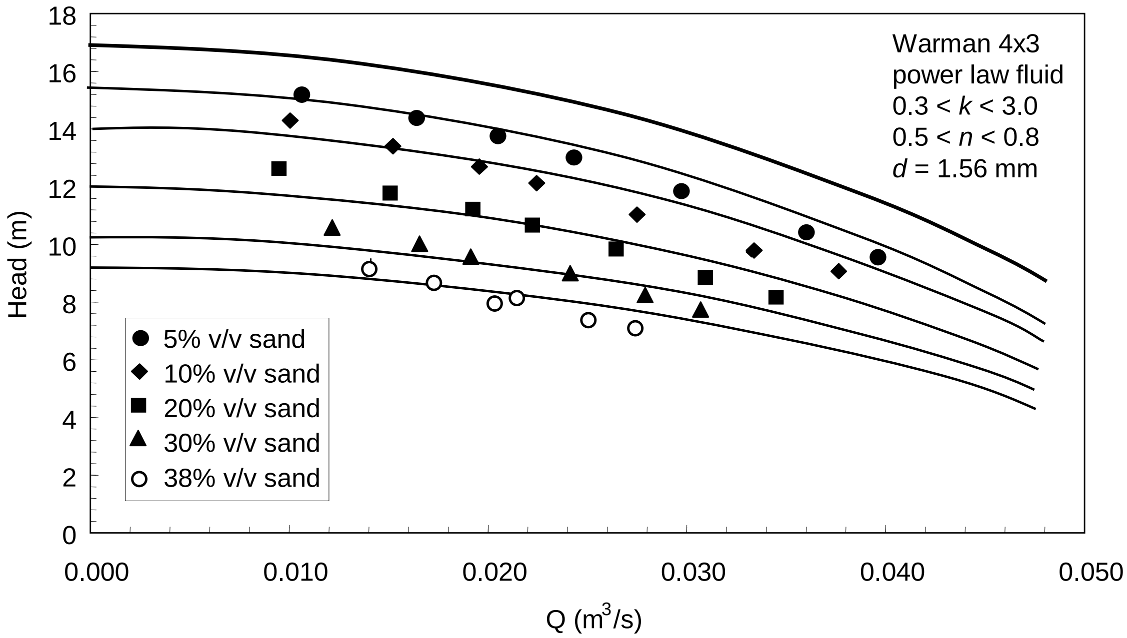

Figure 7 Typical coarse solids de-rating (CMC fluid)

Engin and Gur's correlation, developed for coarse solids in water, based solely on the solid's properties and the pump impeller diameter alone is shown Figure 7 along with the data. The agreement is surprisingly good given the variability of the fluid properties, although it tends to over-predict the data at low flow rates. This is because at these flows the viscosity of the underlying carrier fluid is higher reducing the particle/pump interaction and hence reducing the coarse solids-deration. Sellgren et al. (1999) produced a correlation which agrees substantially with Engin and Gur's correlation at high flow rates but produces less de-rating at lower flow rates as observed here, but unfortunately the variation in the carrier fluids viscosity throughout the tests report here prevents direct comparison with this correlation from being made.

4 CONCLUSIONS

Some findings from pump tests using non-Newtonian fluids and suspensions have been presented. From these studies the following conclusions can be drawn:

- For moderate to high viscosity suspensions typical of paste lines, the flow regime inside a centrifugal pump's passages is most probably laminar.
- A method to estimate the representative viscosity within the pump based on the flow through an equivalent "pipe" passage has been developed. This viscosity is then used in conjunction with the Hydraulic Institute method to estimate the viscous de-rating.
- The exact dimension of the equivalent pipe is left to a fitting exercise for a given series of pumps, and as such is not necessarily a universal solution although the present tests suggest that the characteristic dimension (w) may well be a constant fraction of the pump impeller diameter. However, such an analysis is relatively simple to do for a given pump series and does provide insight into the pump's behaviour.
- The sensitivity of the method to the local shear rate is used to explain the increased viscous de-rating at low flow rates.
- Centrifugal pump tests with complex suspensions consisting of non-Newtonian fluids and coarse particles showed that the head de-rating was primarily a function of the coarse solids concentration. Adequate predictions of head performance are obtained by calculating the effect of solids de-rating alone.

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NOTATION

D_h	Equivalent hydraulic diameter of pump	(m)
D_{imp}	Diameter of pump impeller	(m)
H	Pump head	(m)
k	Consistency index	(Pa s^n)
n	Flow behaviour index	
Q	Pump flow rate	(m^3 s^{-1})
V	Fluid velocity through pump passage	(m s^{-1})
w	Pump characteristic dimension	(m)
$\dot{\gamma}$	Shear rate	(s^{-1})
η	Apparent viscosity	(Pa s)
τ_y	Fluid yield stress	(Pa)