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DERATION OF CENTRIFUGAL PUMPS FOR NON-NEWTONIAN SLURRIES

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Many experimental studies have reported the deration of various designs of centrifugal pump when pumping viscous liquids, pseudohomogeneous Newtonian slurries and coarse particle settling slurries. The process of derating the performance of a given centrifugal pump at a fixed impeller rotational speed involves the estimation of the pump performance through the reduced head, throughput and efficiency compared with water alone. In 1983, the Hydraulic Institute in the USA provided a method for derating pumps for Newtonian viscous liquids, and the approach has been incorporated into the 2015 ANSI/HI standard. By contrast, while there have also been some significant experimental studies in the last 20 years or so which report the deration of centrifugal slurry pumps when pumping various non-Newtonian slurries, some of which include various approaches to pump deration for this slurry type, a standardised and commonly-accepted method has not yet emerged. This paper reviews some published pump deration methods for non-Newtonian slurries.

KEY WORDS: centrifugal pump, non-Newtonian, slurries, derating methods, performance

NOTATION

C_E	correction factor for pump efficiency, -
C_H	correction factor for differential head, -
C_Q	correction factor for flowrate, -
D	internal pipe diameter, m
D_h	hydraulic diameter, m
D_{imp}	pump impeller diameter, m
H	differential head at normal flowrate Q through pump, m
H_s	differential head for slurry flow, m
H_w	differential head for water flow at same pump speed as slurry, m
k_s	proportionality constant between averaged shear rate and agitator/pump speed, -
K	consistency coefficient in Herschel-Bulkley flow model, Pa s^n
Q	volume flowrate, l/s
Q_s	slurry volume flowrate, l/s
Q_w	water volume flowrate, l/s
n'	local flow behaviour index, -
n	flow behaviour index in Herschel-Bulkley flow model, -
N	agitator or pump impeller speed, s^{-1}
V	pipe velocity, m/s
w	characteristic length of pump internal in Pullum et al method, m

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$\dot{\gamma}$	shear rate, s^{-1}
$\dot{\gamma}_b$	boundary shear rate, s^{-1}
η_{app}	apparent viscosity, Pa s
η_s	pump efficiency when pumping slurry, -
η_p	pump efficiency when pumping water, -
ν	slurry kinematic viscosity, cSt
ρ	slurry density, kg/m^3
τ_y	yield stress parameter in Herschel-Bulkley flow model, Pa

1. INTRODUCTION

The process of derating the performance of a centrifugal pump at a fixed impeller rotational speed to allow for pumping a more viscous material than water involves the estimation of the reduced pump performance. This involves head, throughput and efficiency reduction compared with pumping water alone using three deration factors: $C_H = H_s/H_w$, $C_Q = Q_s/Q_w$ and $C_E = \eta_s/\eta_w$ (Xylem, 2012) respectively.

It is impossible to determine a given centrifugal pump performance curve by calculation because of the many design and manufacturing parameters (blade angle, gap width, surface roughness, etc). Therefore, centrifugal pump performance curves are always established experimentally by actual measurement, with water as the standard test liquid. In many cases, however, the fluid pumped (such as slurries) is more viscous. In these cases, pump performance can differ considerably from that when pumping water. Viscous liquids cause more hydraulic losses in the pump so that at higher viscosities pumping head and pump efficiency decrease while required power increases. The pumping head and pump efficiency curves for viscous liquids fall below the corresponding water performance curves, but the shut-off head point remains the same, regardless of viscosity.

In 1983 the Hydraulic Institute in the USA provided a method for derating centrifugal pumps for Newtonian fluids. This became an ISO standard 9906 in 1999 and has subsequently been revised in 2012. Davidson and Bertele (2000) gave an alternative method and there is also the ANSI / HI standard (2015). In the case of coarse particle settling slurries, experimental studies have led to the development of a number of empirically-based equations and nomograms, and to the ANSI / HI standard (2016). While there have been experimental studies using non-Newtonian slurries on centrifugal pump deration, some of which include recommendations for pump deration, there is currently no standardised and commonly-accepted approach that could be applied to non-Newtonian fluids. The three main approaches to pump deration are presented below and have been reviewed recently by Buratto et al (2017). Crawford et al (2012), Furlan et al (2013, 2014, 2016), Mrinal et al (2016), Sellgren et al (2017), Wilson and Sellgren (2006), and Xu et al (2002) have also discussed pump deration for non-Newtonian fluids.

2. LITERATURE REVIEW

It is well-known that when slurries are conveyed using a centrifugal pump the head, throughput and pump efficiency are all reduced compared with that for pumping water alone. For pseudohomogeneous Newtonian or non-Newtonian slurries and also for coarse particle, settling slurries, the effect of a progressively-increasing solids concentration, C_v, on differential head, efficiency and throughput/capacity is shown in Figure 1.

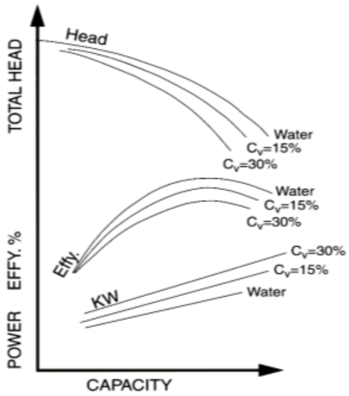

Figure 1. Effect of solids concentration, C_v, on centrifugal pump performance

2.1 PUMP DERATION FOR NEWTONIAN SLURRIES

The HI Chart (1983) shown in Figure 2 can be used to predict pump performance for Newtonian materials of known viscosity when the pump performance for water is known.

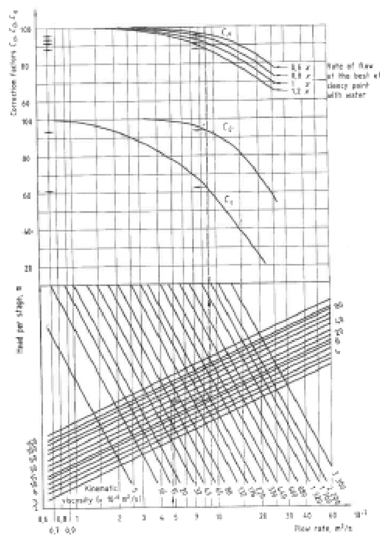

Figure 2. HI chart for derating pump performance for Newtonian materials (Hydraulic Institute, 1983; ISO 9906, 2012)

Guidelines for the effect of viscosity on pump performance have been given as follows (Davidson and Bertele, 2000). Pump capacity, Q, and differential head (the head created by the pump operation), H, are not significantly affected by viscosity provided

$$\nu < (Q)^{0.5} H^{0.25} \qquad (1)$$

The general useful range of centrifugal pumps is limited by:

$$\nu < 100(Q)^{0.5}H^{0.25} \quad (2)$$

Thus if inequality (2) is not satisfied, a positive displacement pump may be preferable. The HI chart was modified by Davidson and Bertele (2000) to give Figure 3, using a parameter, $\nu/(Q^{0.5}H^{0.25})$ and involving lines of best peak efficiency for water pumping.

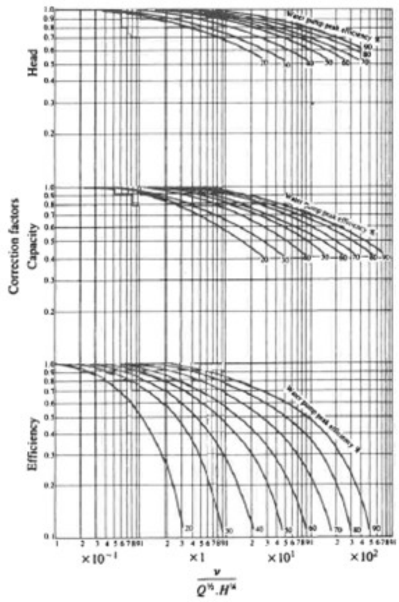

Figure 3. Modified HI chart for derating pump performance for Newtonian materials (Davidson and Bertele, 2000)

The HI approach was developed further in the ANSI / HI standard (2015). However, it is not clear whether Figures 2 and 3 and the ANSI / HI standard provide similar results.

2.2 PUMP DERATION FOR NON-NEWTONIAN SLURRIES

Slurries frequently exhibit a shear-thinning rheological property which is usually modelled using the power law, Bingham plastic or Herschel-Bulkley model (Brown and Heywood, 1991). This can cause a dip in the pump characteristic at low flowrates (Walker and Goulas, 1984; Wilson and Sellgren, 2006; Pullum et al, 2007b) as shown in Figure 4 and can lead to multiple operating (duty) points which are known to occur in practice. Wilson and Sellgren (2006) have also noted that the pump characteristic can show a very slow rise with increasing flowrate. As a result, the pump characteristic and system curve often intercept at a rather shallow angle. The result is that even a small head reduction can result in a disproportionately large reduction in flowrate.

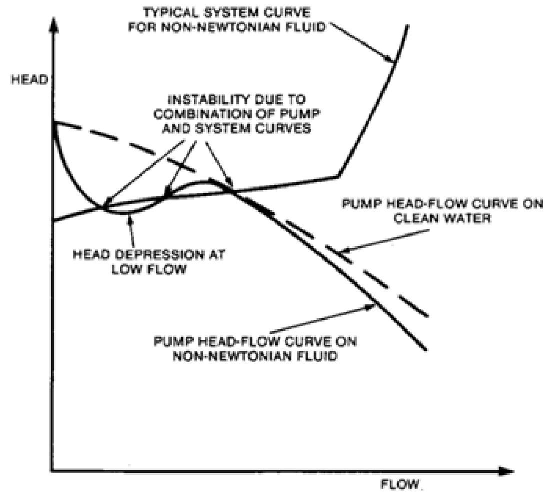

Figure 4. Effect of fluid non-Newtonian property on pump characteristics

There are currently three main published methods for derating head, throughput and efficiency for non-Newtonian fluids including slurries, none of which have yet been accepted as a standard to complement the ANSI / HI standard (2015).

2.2.1 WALKER AND GOULAS METHOD

Walker and Goulas (1984) have investigated the prediction of the performance of centrifugal pumps for non-Newtonian slurries based on the HI Chart. For high flow rates, the plastic viscosity is determined at the highest shear rate possible by fitting a straight line to the material's rheogram above a shear rate of 10% of BEP flow. This was then used as the plastic viscosity in the Bingham plastic model for use in the HI method (1983) to predict pump performance. For low flow rates, an apparent viscosity has to be determined for shear rates corresponding to 2N, where N is the impeller angular speed. They claimed that at low shear rates the apparent viscosity is most sensitive to the influence of the yield stress which appeared to be the dominating factor in the head reduction. Using coal/water and kaolin/water slurries and two different pumps they predicted most data points to within ±5%. However, other studies (Kabamba, 2006; Kalombo, 2013; Sery and Slatter, 2002, 2004) have not been able to achieve similar results, and the reasons for this disagreement are not obvious (Haldenwang et al, 2013).

Although not explicitly mentioned by Walker and Goulas, the existence of a boundary shear rate, $\dot{\gamma}_b$, is advocated. Slatter (2005) confirmed that the boundary shear rate marks the limit between yield stress and plastic viscosity domination of viscosity. However, the calculation of a plastic viscosity for shear rate values of 10 to 1500 s^{-1} makes the procedure of Walker and Goulas a guess-work, or at best a procedure which is valid only when experimental data are available which is contrary to the objective of predicting the pump performance. Furthermore, the plastic viscosity is essentially a fitting parameter with no rheological meaning. It is not applicable to any other rheological model.

2.2.2 SERY ET AL METHOD

Sery et al (2002; 2004; 2005; 2006a,b,c) considered the two methods for estimating the relevant viscosity at either low or high flow rates by Walker and Goulas to be ambiguous. Instead their work involves the calculation of an apparent viscosity from a pump impeller average shear rate determined using the Metzner and Otto (1957) method. This method was originally developed to define a torque-averaged shear rate for the rotation of an agitator in a mixing vessel as k_sN, where N is the impeller speed and k_s has to be found experimentally. k_s is a function of the impeller design but not of the fluid properties. Sery et al determined a characteristic pump impeller average shear rate, using the Metzner and Otto method, to calculate an apparent viscosity based on the fluid's rheology. This was then used with the HI Chart to predict pump deration. The Herschel-Bulkley flow model was used and is given by

$$\tau = \tau_y + K\dot{\gamma}^n \qquad (3)$$

Using the model parameters for different materials a definition of the apparent viscosity was used (Sery et al, 2006) with the HI charts (1983, 1999) to predict head and efficiency for a pump:

$$\eta_{app} = \frac{\tau_y}{k_s N} + K(k_s N)^{n-1} \qquad (4)$$

Xylem (2012) suggests that for sewage sludges a reasonable value for k_s is 4.
A closed 5-vane (type AH) Warman 6/4 centrifugal pump was used in their test work. The impeller is metal-lined with a diameter of 365mm. Pump tests were run using two kaolin slurry concentrations of 21% and 17% by volume, at pump speeds of 1100 and 1400 rpm. Pump suction and discharge pressures were measured with point pressure transducers. Pipe pressure gradient was measured with differential pressure transducers and flow rate using a magnetic flowmeter. A speed-torque unit was fitted between the motor and the pump, so that the mechanical drive power could be determined.

The performance prediction of the 6/4 Warman centrifugal pump based on the HI Chart was examined. Test data were compared with predicted values from the HI Chart and it is suggested that as far as the head is concerned, the modified Metzner and Otto method could yield results with a reasonable accuracy (less than 5%). However, in the 60% to 120% BEP (best efficiency point) region and where data are available, the error in predicted efficiency was as high as 20%. It was also observed that the error margin is decreased as the fluid becomes more viscous. A plausible explanation for this is that their relationship between the impeller average shear rate and pump speed applies only when viscous forces dominate the process under investigation (laminar regime). The Metzner-Otto method for agitators is valid only when the agitator Reynolds number is less than 10, and so within the laminar region. Pump power requirements would be underestimated when the turbulent flow regime prevails. This raises the concept of a boundary shear rate or critical Reynolds number which perhaps needs to be defined.

2.2.3 PULLUM ET AL METHOD

Pullum et al (2007a, b; 2011) and Graham et al (2009) recognised that the viscosity is not constant but a function of the local shear rate in the pump's volute. A representative shear rate for the flow through the pump is defined, and therefore the representative shear stress, which is then taken from the measured material's rheogram. They defined an equivalent hydraulic pipe diameter, D_h, based on the pump dimensions. The method requires that a characteristic dimension, w, which varies for a specific pump design is determined, and D_h is then used to determine the velocity through the equivalent hydraulic pipe. The representative shear rate is then derived from both the velocity and pipe diameter and is used to obtain the representative shear viscosity. This was used with the HI method, but could be used in the Davidson and Bertele (2000) and the ANSI / HI standard (2015). The flow through the rotor passage (called the "equivalent hydraulic pipe") is assumed to be laminar. The hydraulic diameter of this "pipe", D_h, is defined by

$$D_h = \frac{4w\pi D_{imp}}{2(\pi D_{imp} + w)} \tag{5}$$

It has been suggested the parameter "w" is the rotor passage width, but it is very unlikely that such a value would be correct because the equivalent pipe must account for all flows within the pump and so is fictitious (Graham et al, 2009), although it has been defined as the "width between the shrouds of a closed impeller" (Sellgren et al, 2017). Values of 0.24 and $0.27D_{imp}$ have been used to fit test data from two pumps to within ±10% (Graham et al, 2009). The velocity in the pipe is then calculated from

$$V = \frac{4(\frac{Q}{1000})}{\pi D_h^2} \tag{6}$$

For laminar flow in the "pipe", the true wall shear rate is obtained using the Rabinowitsch-Mooney equation:

$$\dot{\gamma} = \left[\frac{1+3n'}{4n'}\right]\left[\frac{8V}{D_h}\right] \tag{7}$$

This is used to obtain the apparent viscosity from the material's rheogram for use with one of the three deration methods for Newtonian slurries.

Buratto et al (2017) used a wide range of fluids in their experimental work and found that the parameter "w" varies with the fluid used from 3.1% to 7.9% of D_{imp} with one pump type, and 3.5% to 37.5% of D_{imp} for a second pump. They suggest this is in contrast to the method reported by Pullum et al (2007b) which uses the same value of "w" for a given pump for all fluids used. Furlan et al (2013, 2014, 2016) found values of "w" of 13.7% and 15.8% of D_{imp} for a metal-lined 3-vane, 0.31m diameter pump at 1450 and 1800 rpm respectively. Kalombo et al (2014) suggest that since Pullum et al (2007b) used their own data to estimate the best value of "w" which, in turn was used to predict their own experiments, this shows more of their goodness of fit, and as such it does not constitute an independent validation of their approach. It is therefore not possible to predict the value of "w" *a priori* for a new application and therefore must remain an empirical approach. Kalombo et al (2014) also noted that the Pullum et al approach failed to predict the pump efficiency when pumping 21% to 28% kaolin slurries and 5% to 9% aqueous CMC mixtures and so the apparent viscosity used in this approach cannot be the only representative non-Newtonian viscosity to be used in the HI method. However, it did predict the pump head better than the Walker and Goulas method (1984).

3. CONCLUSIONS

Current approaches try to establish a single slurry viscosity based on the slurry flow curve using a single representative shear rate for the normally wide range of shear rates occurring between the pump impeller and the casing. Using this single viscosity, estimates of the three correction factors are then made using the HI Chart (1983) or its variants (ISO, 2012; Davidson and Bertele, 2000; ANSI / HI, 2015). These methods based on Newtonian viscous liquids are assumed applicable to non-Newtonian slurries. However, perhaps this is not the case and fresh approaches to pump deration for non-Newtonian slurries should be taken which may involve more than a single representative viscosity, allowing for the wide range of shear rates which occurs in the pump casing.

REFERENCES

1. ANSI / Hydraulic Institute. 2015. Rotodynamic Pumps – Guideline for Effects of Liquid Viscosity Performance (ANSI/HI 9.6.7 – 2015), New Jersey, USA. (www.pumps.org)
2. ANSI / Hydraulic Institute. 2016. Rotodynamic (Centrifugal) Slurry Pumps for Nomenclature, Definitions, and Operation (ANSI/HI 12.1 – 12.6 – 2016), New Jersey, USA. (www.pumps.org)
3. Brown. P., Heywood, N.I., 1991. Slurry Handling: Design of Solid-Liquid Systems. Springer.
4. Buratto, C., Occari, M., Aldi, N., Casari, N., Pinelli, M., Spina, P.R., Suman, A. 2017. Centrifugal pumps performance estimation with non-Newtonian fluids: review and critical analysis. Proc. 12th European Conf.on Turbomachinery Fluid Dynamics and Thermodynamics, Stockholm, Sweden, 3 - 7 April 2017. ETC12. Paper ID: ETC2017-248.
5. Crawford, J., van Sittert, E., van der Walt, M., 2012. The performance of centrifugal pumps when pumping ultra-viscous paste slurries. Journal of the Southern African Institute of Mining and Metallurgy, 112, 959-964.
6. Davidson, J., von Bertele, O., 2000. Process Pump Selection: A Systems Approach, 2nd Edition. Instn Mechanical Engrs, London,UK, p. 88.
7. Furlan, J., Visintainer, R., Sellgren, A., 2013. Pumping a 100 to 600 Pa yield stress clay slurry highly non-Newtonian clays and tailings slurries. Proc. Int. Seminar on Paste and Thickened Tailings (Paste 2006), 17-20 June 2013, Belo Horizonte, Brazil.
8. Furlan, J., Visintainer, R., Sellgren, A., 2014. Centrifugal pump performance when handling highly non-Newtonian clays and tailings slurries. Proc. Hydrotransport 19, Golden, Colorado, USA, pp 117-130. BHR Group, Cranfield, UK
9. Furlan, J., Visintainer, R., Sellgren, A., 2016 Centrifugal pump performance when handling highly non-Newtonian clays and tailings. Can J Chem Engng, 94, 1108-1115.
10. Graham, L.J.W., Pullum, L., Slatter, P.T., Sery, G., Rudman, M., 2009. Centrifugal pump performance calculation for homogeneous suspensions. Can J Chem Engng, 87(4), 526-533.
11. Haldenwang, R., Kalombo, J.-J., Chhabra, R., 2013. Prediction of centrifugal pump and head when pumping non-Newtonian sludges. Proc. 16th Int. Conf. on Transport and Sedimentation of Particles, 18 - 23 Sept. 2013, Rostock, Germany, pp 9-18.
12. Hydraulic Institute. 1983. Hydraulic Institute Standards for Centrifugal, Rotary and Reciprocating Pumps. 14th edn., Cleveland, Ohio, USA.
13. ISO 9906: 2012. Rotodynamic Pumps – Hydraulic Performance Acceptance Tests.
14. Kabamba, B.M., 2006. Evaluation of Derating Procedures for Centrifugal Pump Performance,

15. Magister Technologiae thesis, Cape Peninsula University of Technology, South Africa.
16. Kalombo, J.J.N., 2013. Centrifugal Pump Derating for non-Newtonian Slurries: Analysis of the Viscosity to be used in the Hydraulic Institute Method. Magister Technologiae thesis, Cape Peninsula University of Technology, South Africa.
17. Kalombo, J.J.N., Haldenwang, R., Chhabra, R.P., Fester, V.G., 2014. Centrifugal pump derating for Non-Newtonian slurries. J Fluids Engng, March, 136, 031302-1 to -11.
18. Metzner, A.B, Otto, R.E., 1957. Agitation of non-Newtonian fluids. AIChEJ, 3, 3-10.
19. Mrinal, K.R., Hamid Saddique, Md. Abdus Samad 2016. Performance prediction of a centrifugal pump delivering non-Newtonian slurry, Particulate Science and Technology, 36, 38-45.
20. Pullum, L., Graham, L.J.W., Rudman, M., 2007a. A de-rating method for centrifugal pumps pumping "paste". Proc. 10th Int. Seminar on Paste and Thickened Tailings conference (Paste 2007), Fremantle, Australia.
21. Pullum, L., Graham, L.G.W., Rudman, M., 2007b. Centrifugal pump performance calculation for homogeneous and complex heterogeneous suspensions, Journal of the Southern African Institute of Mining and Metallurgy, 107 (6), June, 373-379.
22. Pullum, L, Graham, L.J.W., Wu, J., 2011. Centrifugal pump performance with non-Newtonian slurries. Proc. 15th Int. Conf. on Transport and Sedimentation of Particles, Wroclaw, Poland, pp 115-129.
23. Sellgren, A., Visitainer, R., Furian, J. 2017. Centrifugal slurry pump performance deratings - a coherent approach. Proc. Hydrotransport 20, Melbourne, Australia, pp 401-414. BHR Group, Cranfield, UK
24. Sery, G.A., Slatter, P.T., 2002. Centrifugal pump derating for non-Newtonian slurries. Proc. Hydrotransport 15, Banff, Canada, 679-692, BHR Group, Cranfield, UK.
25. Sery, G.A. Slatter, P.T., 2004. Centrifugal pump performance Reynolds number for non-Newtonian slurries, Proc 12th Int. Conf. on Transport and Sedimentation of Solid Particles, Prague, Czech Republic, 20-24 September 2004, ISBN 80-239-3465-1, pp 601-609.
26. Sery, G.A., Kabamba, B.M., Slatter, P.T., 2005. Evaluation of derating procedures for centrifugal pump slurry service, Proc. MINPROC '05, SAIMM Western Cape, 4-5 August 2005, Lord Charles Hotel, Somerset West, South Africa.
27. Sery, G.A., Slatter, P.T. 2006a. Centrifugal pump derating using a mixing power approach, Proc. MINPROC '06, SAIMM Western Cape, 3-4 August 2006, Cape Town, RSA.
28. Sery, G.A., Slatter, P.T., 2006b. Centrifugal pump viscometry, Proc. SASOR 2006, 1st Southern African Conference on Rheology, 24-27 Sept. 2006, Cape Town, RSA.
29. Sery, G.A., Kabamba, B.M., Slatter, P.T., 2006c. Paste pumping with centrifugal pumps: evaluation of the Hydraulic Institute chart de-rating procedures, Proc. 9th Int. Seminar on Paste and Thickened Tailings (Paste 2006), Limerick, Ireland, pp 403-412.
30. Slatter, P.T., 2005. Tailings transport – back to basics! Proc. 8th Int. Seminar on Paste and Thickened Tailings (Paste 2005), Santiago, Chile, 20-22 April, pp 165-176.
31. Walker, C.I., Goulas, A., 1984 Performance characteristics of centrifugal pumps when handling non-Newtonian homogeneous slurries. Proc. I Mech E, 198(A), 41-49.
32. Wilson, K.C., Sellgren, A., 2006. Effect of strongly non-Newtonian slurries on the head and efficiency of centrifugal pumps, Proc. 11th Int. Conf. on Transport and Sedimentation of Solid Particles, Georgia, September, pp 342-348.
33. Xu, J., Tipman, R., Gillies, R., Shook, C.A., 2002. Centrifugal pump performance with Newtonian and non-Newtonian slurries. Proc. Hydrotransport 15, Banff, Canada, pp 693-709.
34. Xylem (2012) Handbook of Sludge Pumping. Xylem Water Solutions AB. Sundbyberg, Sweden.