The new ANSI/HI centrifugal slurry pump standard

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Transaction Paper

Synopsis

The Hydraulic Institute has completed the task of developing a new ANSI/HI standard 12.1-12.6 (2005) for rotodynamic (centrifugal) slurry pumps covering nomenclature, definitions, applications, and operation. The standard provides examples of the different slurry pump types and contains an extensive section on pump and slurry definitions. The effect of slurry on pump performance is covered along with the pumping of froth. Reference is also made to ANSI/HI standard 9.6.7 (2004), which contains a new method for pump performance correction when handling viscous fluids.

Classification of slurry services is established and then is used to determine limitations on velocities and total head per pump in order to obtain acceptable wear performance. The new service class, head per stage and other limits are directly related to capital and other cost considerations that will affect solids transport system economics. The writers review the contents of the new standard, highlight the main points, and discuss the reason for the slurry classification, corresponding limits and expected implications, particularly with respect to operating costs of the pumps in solids transport systems.

Introduction

The new standard was developed by a special Slurry Pump Standard Committee formed by the Hydraulic Institute and was issued in conjunction with the American National Standards Institute. Senior engineering personnel from almost all of the slurry pumps manufacturers participating in the Hydraulic Institute worked on the new standard. The process took over five years to complete.

More than ten pump manufacturers took part in the development, along with user companies and industrial and academic experts. The two first authors were permanent members of the committee, while the third was a major academic contributor.

Format of the standard

The standard follows the usual HI format that starts with a scope, pump types and nomenclature. Figure 1 below shows typical materials used, dependent pressure and particle size.

The document covers all types of centrifugal pumps used for pumping slurries, such as: separately coupled frame mounted, close coupled, horizontal, wet pit cantilver, end suction submersible, lined, unlined, metal and elastomer pump types. Examples are given of the different types of slurry pumps available along with parts lists, and definitions. (See Figure 2.)

Terms and definitions

As a unique HI standard based more on application rather than a specific pump type, the document contains an extensive section on slurry service terms and definitions. Charts are provided for transport rate conversion into different pipeline velocities, weight to volume conversions, and for flow regions as a guide to a slurry being settling or non-settling. All charts are provided in dual units. Settling (heterogeneous) and non-settling (homogeneous) pipeline pumping characteristics are described schematically. A rough basis for classification is given in Figure 3.

Effect of slurry on pump performance

The performance of a centrifugal pump is affected by the solids in the slurry. Slurries with most of the particles smaller than about 75 μm showing a non-settling viscous behaviour are considered in the subsequent section.

Settling slurries

The pump performance derating, or 'solids effect' for settling slurries is normally described in terms of the ratios and factors given in Figure 4.

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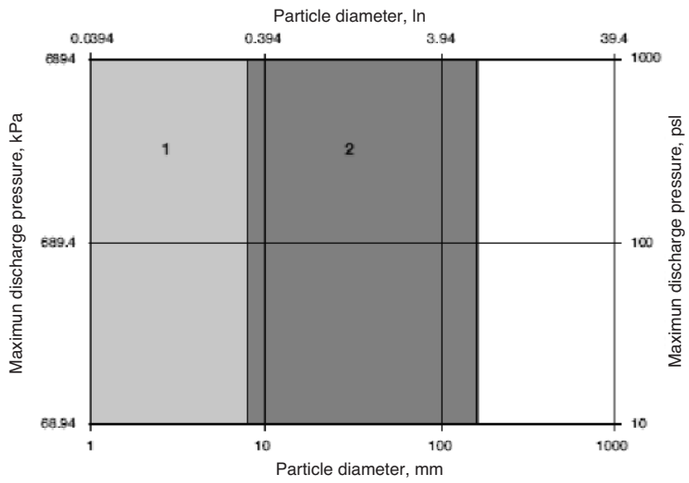
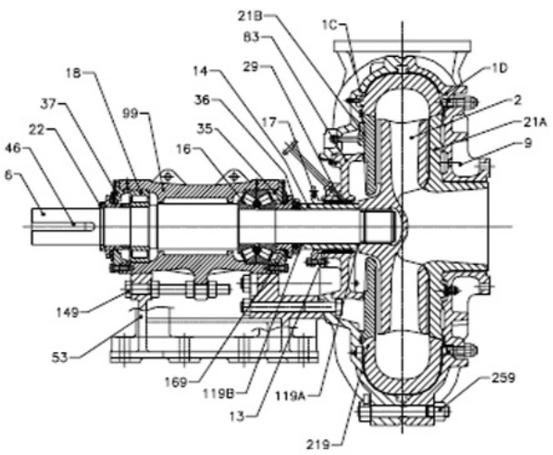


Figure 1—Typical material types for discharge pressure and particle size



1C	Casing, gland half	1D	Casing, suction half
2	Impeller	6	Shaft
9	Cover, suction	13	Packing
14	Sleeve, shaft	16	Bearing, inboard
17	Gland	18	Bearing, outboard
21A	Liner, suction cover	21B	Liner, stuffing box cover
22	Locknut, bearing	29	Ring, lantern
35	Cover, bearing, inboard	36	Collar, release
37	Cover, bearing, outboard	46	Key, coupling
53	Base	67	Shim
83	Stuffing box	99	Housing, bearing
119A	O-ring	119B	O-ring
149	Screw, impeller adjusting	169	Seal, bearing housing
219	Liner, casing		

Figure 2—Typical slurry pump section and parts list

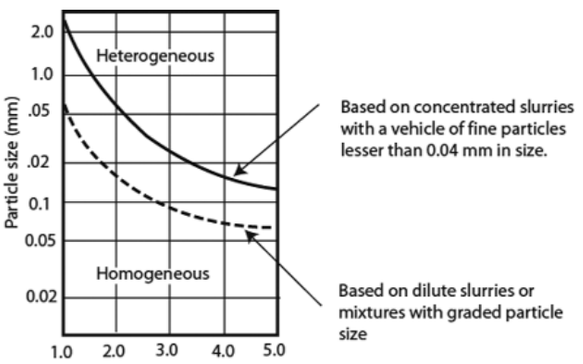


Figure 3—Schematic classification of slurry flows for various solid density ratios (After Aude *et al.*)

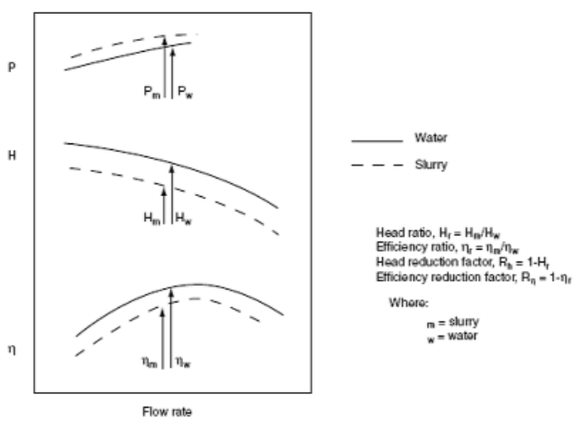


Figure 4—Effect of slurry on pump characteristics (schematic)

The factors and ratios defined in Figure 4 are based on the assumption that there is no dependence on flow rate or rotary speed. Within normal operating ranges this is usually a reasonable assumption.

Concentrating on leading parameters, a generalized design diagram for estimation of the performance derating is given in Figure 5. The diagram gives the head reduction factor RH in terms of pump impeller diameter (D) and

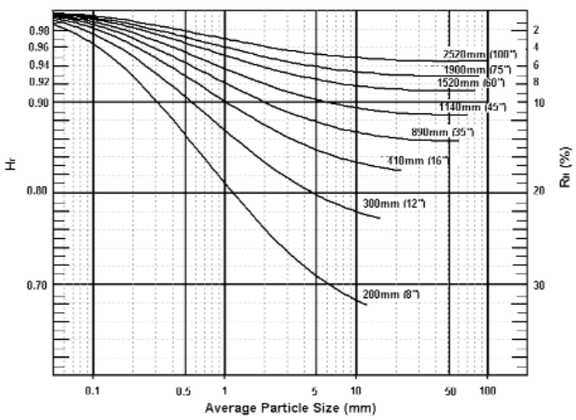


Figure 5—Generalized solids-effect diagram for pumps of various sizes (impeller diameters). For solids concentration by volume, C_v = 15% with relative density of solids, S_s = 2.65 and a negligible amount of fine particles, X_h

average solid particle size (d_{50}) for a baseline or reference slurry with 15% by volume solids of 2.65 relative density and a negligible amount of fine particles. R_H values obtained from Figure 5 are concurrently multiplied by correction factors C_{CV}, C_S, and C_{fp} for slurries with different concentration (C_V), different relative density of solids (S_S), and different content of fine (less than 75 micron) particles (X_h), respectively. Equations for these correction factors and a brief example follow.

$$C_{CV} = C_V / 15$$
$$C_S = \left[\frac{S_S - 1}{1.65}\right]^{0.65} \qquad [1]$$
$$C_{fp} = \left(1 - X_h\right)^2$$

For power, it is assumed that the $R_h = R_\eta$ which means that the power consumption increases directly with the slurry density, i.e. $P_m = S_m P_w$ where S_m is the slurry density and the powers P_m and P_w were defined in Figure 4. This assumption is sometimes conservative for large pumps. With small pumps and slurries of well over about 20% volumetric concentration, the power may be larger than expressed above, dependent on the individual properties of the solids.

Example

A pump with a 0.89 m diameter impeller is used at C_v 15% for a 2.65 specific gravity slurry of average size 1 mm with no fine particles, will according to Figure 5 have an R_h of about 8% (H_r = 0.92). If the per cent of fine particles less than 75 micron (X_h) is increased from zero to 20%, then C_{fp} = (1-0.2)^2 = 0.64 giving an R_h about 5% instead of 8%.

Non-settling slurries

Viscous liquids

In accordance with the known Hydraulic Institute procedure, the change in pump performance when pumping viscous Newtonian liquids for given water curves and rotary speed is expressed by correction factors for head, C_H, flow rate, C_Q, and efficiency, C_η. In an extension of the previous HI procedure for viscosity corrections, ANSI/HI 9.6.7 (2004) provides a generalized derating method linked to a large body of experimental data and based on a pump performance Reynolds number adjusted for specific speed.

The leading parameter, B, in the method is coupled to various empirical relationships for the correction factors which also are represented graphically, see Equation [2] and Figure 6 where the correction factors also are defined. C_H and C_Q express the viscous fluid head over the water head and the viscous fluid flow rate over the water flow rate, respectively. C_η expresses the efficiency in viscous service at the reduced flow rate over the efficiency for the unaffected flow rate in water service. B is expressed as follows:

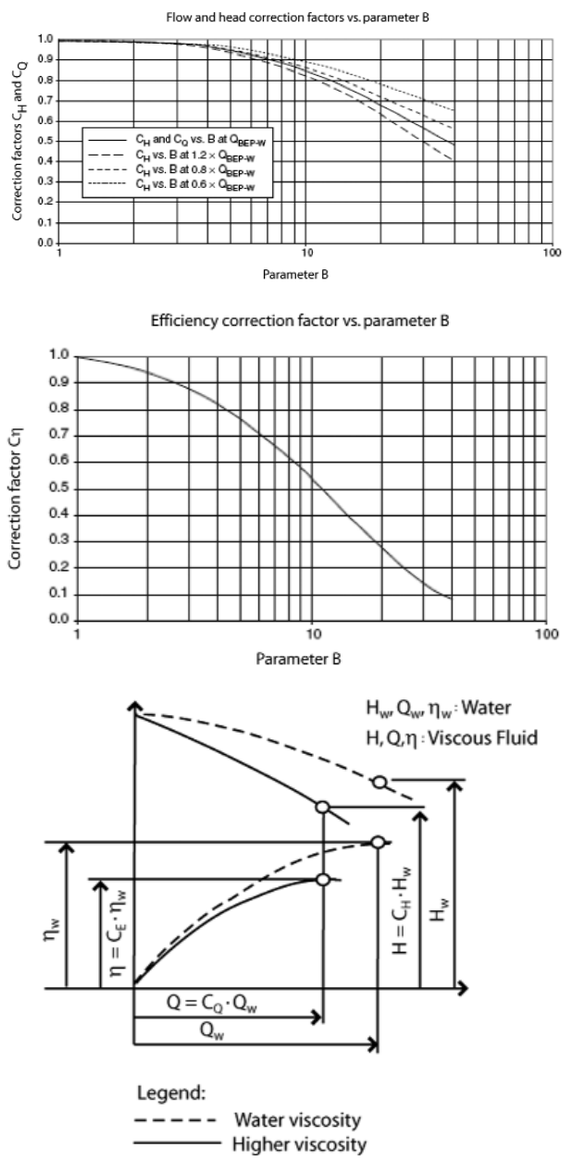


Figure 6—Schematic representation of the charts coupled to the parameter *B* in Equation [2], when pumping a viscous Newtonian liquid, ANSI/HI (2004). The corresponding correction factors are defined and related to water pumping in the sketch

$$B = \frac{16.5 v^{0.5} H^{0.0625}}{N^{0.25} Q^{0.375}} \quad [2]$$

where v is in cSt , Q in m³/h, H in m and N in rpm. It follows from Equation [2] that large flow rates, i.e. large pumps, are affected less than smaller pumps and that higher rotary speeds reduce the effect for a given head and viscosity.

Slurries

The HI method is an approximate, but reasonable representation of the effect of Newtonian liquids on the performance for most practical purposes for kinematic Newtonian viscosities less than about 4 000 times that for water, ANSI/HI (2004). It is also stated that effects related to non-Newtonian liquids and particle-liquid mixtures, i.e. slurries, are beyond the scope of the method. Performance tests should be carried out with the selected pump and particular viscous medium to be handled, when accurate information is essential.

With non-Newtonian fluids and slurries, pump performance effects can vary widely. Highly-concentrated slurries of fine particles are, for practical pumping purposes, non-settling and behave in a highly non-Newtonian way and often exhibit a yield stress. The yield stress is defined as the minimum stress required to cause the slurry to flow.

Most reported experimental pump performance results for highly non-Newtonian slurries are for kaolin where particles normally are smaller than about 2 μm. In the light of the difficulties expressed above there are reported results which have been related to a characteristic viscosity for use in Equation [2].

Highly concentrated industrial slurries can often be approximately represented by a yield stress and a straight line for larger shear strains in a rheogram. The slope of the line is often denoted 'tangent viscosity'. If the linear relationship prevails to zero shear strain then the slurry is a Bingham type of medium, where the slope (tangent viscosity) is termed plastic viscosity or coefficient of rigidity. These and other viscosity concepts have been employed to characterize a slurry viscosity for use in the HI method to obtain the correction coefficients from Figure 6 for non-settling non-Newtonian slurries.

Pumping froth

The transfer of froths with centrifugal slurry pumps is a special purpose application commonly encountered in the launders of flotation circuits. Presence of entrained air or gas in slurry at the suction inlet decreases the head, flow and efficiency of a pump and, with increasing amounts of air, the losses will also increase. The very large proportion of air in the froth being handled upsets the normal relationships that are used to predict pumping performance and requires a unique approach in selecting and applying pumps for this service.

The standard gives a detailed description of the application of a 'froth factor' for pump selection and sizing. The froth factor is a multiplier that increases the process design capacity to allow for the increased passing volume caused by the gas in the froth. The factored volume usually causes the pump to be at least one pipe size larger than would normally be selected.

Over sizing the pump increases its ability to handle multiphase mixture containing air or gas due to the following reasons:

- The larger inlet diameter provides for more physical space to cope with an accumulation of air without restricting the fluid flow into the pump.
- The increased impeller eye diameter requires a suction pipe of larger diameter. This leads to reduction in the suction line velocity, thus reducing line losses, and therefore increases the available NPSH of the system.
- The impeller of larger diameter will run at a lower speed of rotation, which will result in reduction of the NPSH required by the pump.

Depending on the manufacturer and the application, the factor applied will vary; typical values are 1.5 to 4 but could be as high as eight. Many factors influence the size of the froth factor and these may include the viscosity of the liquid, the size of grind of the mineral, and the chemistry used in the process. The type of pump selected will also have an effect. Some vertical pump froth factors for common processes are given in Table I.

When specifying a pump, it is important that the supplier is made aware of the air free volume being pumped, as well as the established froth factor or volumetric air concentration at the pump suction. For example, if the air free volume required is 500 m³/h and the froth factor has been specified as 2, the total volume pumped can be as high as 500 × 2 = 1 000 m³/h. The pump must be capable of handling this increased rate of flow at the air free specific gravity, since it may be required to pump a fluid without froth on occasions.

It is important to understand that it is the volume of air at the pump inlet that affects the pump performance. Air volumetric concentration directly depends on suction pressure. For this reason the froth pumps should always operate with positive suction pressure, and the higher the pressure, the better the pump performance that can be expected.

Although applying a larger pump may be considered a cost implication, this approach has proven to work well in many processes, especially with vertical pumps, applications of which normally fall into the low system head range with heads seldom over 20 metres. Where higher heads are required, horizontal froth pumps can be used in series. Special increased inlets and suction arrangements, such as flow inducers and specially profiled vanes, proved to be enhancing froth handling.

Table I

Approximate froth factors

Application	Pump froth factor
Copper rougher concentrates	1.5
Copper cleaner concentrates	3.0
Molybdenum rougher concentrates	2.0
Molybdenum concentrates	3.0
Potash	2.0
Iron concentrates	4.0 to 6.0
Coal	6.0

Effect of slurry on pump NPSH required

When pumping slurry, the net positive suction head required by a centrifugal pump in order not to exceed 3% head drop (NPSHR) will increase, in most circumstances, compared to that on water. Effects of solids on NPSH are dependent on the slurry type and the pump design.

For settling slurries of low to medium concentration, a modest increase in NPSHR can be expected. For a particular application this increase can be conservatively estimated by dividing the value of NPSHR on water by the head ratio H_r discussed above.

For viscous and non-settling slurries or mixtures with entrained air, the effect on NPSHR can be significantly greater. The standard suggests that 'the pump manufacturer should be consulted for guidance regarding slurry effects on NPSHR'. Some relevant information can be found in Roudnev (2004).

Pump wear considerations

The new standard provides information on wear mechanisms, wear coefficients, and the response of different materials. The document refers to numerical methods of calculating wear, but stops short of proposing any method. Rather, the standard provides a service class chart shown in Figure 7 for identifying wear severity, a table illustrating corresponding maximum permissible head per stage and impeller velocity limits (see Table II), as well as recommended operating ranges for different designs of discharge casings.

The background information to the head limits recommended in Table II, not provided in the standard, is presented in the following section.

Head limits and costs

The 40 metre head per stage limit comes from the well-proven tip speed limits that have been used over the years in conjunction with a service class chart in Figure 7. The limit established was for acceptable wear, but nowhere was acceptable wear defined in hours of operation or other measure.

Sellgren *et al.* (2005) found that wear rate seemed to be independent of flow for different size similar pumps selected near their design point for a given duty. On the other hand, it was also found that wear and operating cost varied considerably with different head per stage, particle size, and solids concentration duties.

Furthermore, it was shown that Class 4 service wear parts cost approached half of the total operating cost (capital + energy + wear) for the pump, but that (for all services) it remained approximately the same on a cost per ton of solids transported basis regardless of the pump specific speed (N_s) for a given head, solids size, and concentration duty. It was also noted that the higher suction liner wear and slightly higher efficiency of high N_s pumps seemed to be offset by the higher shell wear cost and higher capital cost of the lower N_s pumps.

Taking the calculated wear rate data from Sellgren *et al.* (2005) and assuming a standardized thickness for the limiting section of the component, it is possible to come up with a component life for the impeller, suction liner, and shell wet end parts of the slurry pump.

Component costs vary greatly with the size (flow wise) of the pumps and the volume of solids transported.

Using the life and cost of the pump components divided by the tons of solids transported, it is possible to come up with a unit cost that can be compared with different heads and specific speed designs.

While different pressure ratings and construction styles cause significant data points scatter, a trend and order of magnitude number can be discerned that can be used to estimate wear component consumed cost for a given solids transportation case.

Here it should be noted that it is assumed parts are replaced as they wear with no allowance for downtime or time (labour) to change out parts.

Here it should be noted these are for normalized designs with small impeller shroud clearing vanes and a conventional flat nose face. The shell, impeller and the suction liner are assumed to be in high chrome white iron while the abrading slurry solids are of silica sand.

For solids size other than 300 micron and for different materials, the wear coefficient chart, Figure 12.22 in the standard, can be used to estimate the effect of these variables.

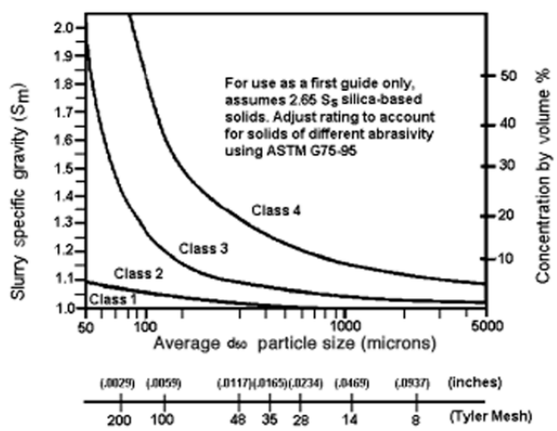


Figure 7—Service class chart for slurry pump erosive wear

Table II

Suggested maximum operating values for acceptable wear

	Service class			
	1	2	3	4
Maximum head per stage:				
metre	123	66	52	40
feet	400	225	168	130
Maximum impeller peripheral speed:				
All-metal pump				
(m/s)	43	38	33	28
(ft/min)	8500	7500	6500	5500
Rubber-lined pump				
(m/s)	31	28	26	23
(ft/min)	6000	5500	5000	4500

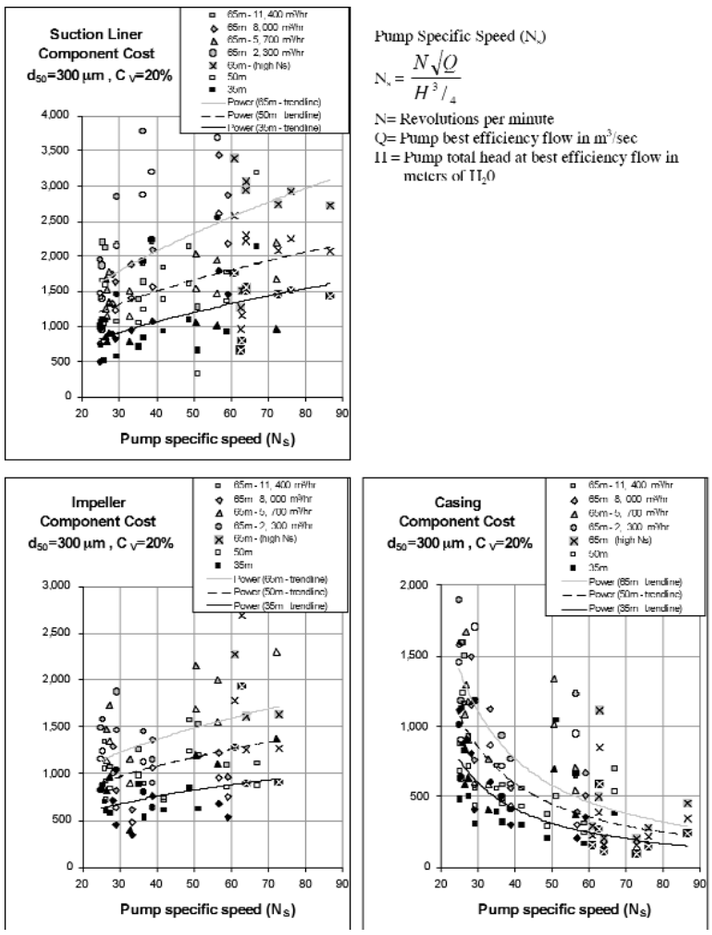


Figure 8—Overall trends of cost for suction liner, impeller, and casing components for various specific speeds and heads independent of pump size

In almost all cases, the suction liner life determines the time between maintenance intervals. If, as in a lot of cases, there are no back-up pumps, then this will incur a downtime cost, which can be significant.

Figure 9 shows the calculated downtime occurrences for three different heads at a solids concentration by volume of 20% with an average particle size of 300 micron at different pump design specific speeds. This shows that larger time intervals between maintenance or lower downtime (where it applies) cost can be achieved either by lower head per stage or lower N_S designs or a combination thereof.

Construction details

The new standard describes wetted materials of construction and gives recommendations for those along with the arrangement details. Recommended bearing housing and rotating shaft seals are described in some detail with a special section on mechanical seals and where they may be used. The standard lays out shaft alignment limits and recommends minimum bearing lives as shown in Table III. Required calculated bearing life is longer for more severe services in recognition of the large impact and unsteady loads encountered.

Installation and other considerations

The standard includes information on a new method of determining maximum permissible flange loads based on the pump hold down bolts as the limiting criteria.

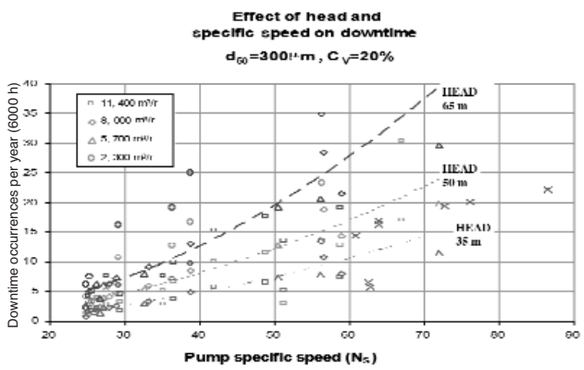


Figure 9—Overall downtime trends per year versus specific speed for various pumping heads

Table III

Calculated fatigue life of bearings by slurry service class

Slurry service class (Ref. Figure 7)	Minimum calculated bearing fatigue life (L_{10} life in hours)
1	17 500
2	35 000
3	50 000
4	50 000 (note 1)

Note 1: For large pumps for class 4 service, it is suggested, life should be increased 2 000 hours for every inch (25.4 mm) of suction diameter over 12 inches (305 mm).
Note 2: Lives noted are for the pump specified operating conditions.

Guideline sections are provided on commissioning, start-up, adjustment, storage, and disassembly. A special section is provided on possible operating problems and maintenance procedures for maximum part life.

Acceptance and other tests are covered along with a brief section on what measurements should be taken and the types of instruments that need to be employed.

The standard includes a sample equipment data sheet that contains information necessary to specify and size the slurry pump, an extensive source material section, a reference section and appendices containing unit conversions, material conversions, and an index.

Conclusion

A new Slurry Pump Standard that has been produced by the Hydraulic Institute is the first HI normative document based on type of service. The standard provides information on centrifugal slurry pump types, nomenclature, definitions, and recommends application limits.

Additional information and the implications of the solids effect, slurry viscosity corrections, the new head limits, and the cost of wear are discussed.

References

1. ANSI/HI 12.1-12.6. Standard for rotodynamic (centrifugal) slurry pumps. 2005.
2. ANSI/HI 9.6.7. Effects of liquid viscosity on rotodynamic (centrifugal and vertical) pump performance. 2004.
3. AUDE, T.C., COWPER, N.T., THOMPSON, T.L., and WASP, E.J. Slurry piping systems: Trends, *Chemical Engineering*, June 1971.
4. ROUDNEV, A.S. Slurry Pump Suction Performance Considerations. *Proc. 16th Intl. Conference on Hydrotransport*, BHR Group, Santiago, Chile, 2004, pp. 13–150.
5. SELLGREN, A., ADDIE, G.A., VISINTAINER, R., and PAGALTHIVARTHI, K. Prediction of slurry pump component wear and cost, *Proceedings, WEDA XXV and Texas A& M Annual Dredging Seminar*, New Orleans, U.S.A., June 2005.
6. WALKER, C.I. and GOULAS, A. Performance characteristics of centrifugal pumps when handling homogeneous slurries. *Proc .Instn. Mech. Engrs*. 1984, no. 1, pp. 41–49.
7. WILSON, K. C., ADDIE, G.R., SELLGREN, A., and CLIFT, R. *Slurry Transport Using Centrifugal Pumps*, 3rd Edn., Springer, New York, USA, 2006, 435 pp. ◆