9.16.2 APPLICATION AND CONSTRUCTION OF CENTRIFUGAL SOLIDS HANDLING PUMPS

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This section discusses the application of centrifugal pumps to the transport of slurries as introduced in Subsection 9.16.1.

Where the distance is large and the size of the solids less than about 100 microns, positive displacement pumps are usually applied. If the solids to be transported are smaller than about 60 mm in size, water is readily available or already part of the process, and the distance is less than 12 miles (20 km), a pipeline driven by centrifugal slurry pumps is usually more cost-effective than belt conveyors, trucks, or railway transportation.

Types of solids transported vary from sag mill feed in a copper mine, iron ore, phosphate matrix, coal, rock salt, tar sands, red mud, various types of waste tailings, and crushed rock and silt. An 18 in (0.45 m) diameter pipeline, for example, is capable of transporting as many as 2200 tons/hr (2000 tonnes/hr) of sand. This equates to about one hundred truckloads per hour over the distance involved.

The application that involves the largest quantities is in the dredging industry, continually maintaining navigation in harbors and rivers, altering coastlines, and mining material for landfill and construction purposes. Because a single dredge may be required to maintain a throughput of 7000 tons/hr of slurry or more, very large centrifugal pumps are used. Figures 1 and 2 show, respectively, an exterior view of a dredge pump on test, and a view of a large slurry pump impeller (Addie and Helmly, 1989).

Because the aim is to transport solids and not water, the higher the concentration of solids, the better for energy consumption and capital cost. This does, however, mean that wear due to the abrasive solids will be significant, requiring a special type of centrifugal pump design and the use of special materials. Slurry pumps may be selected for low-concentration dirty-water service. Most of what follows, however, is about pumps designed and built to give cost-effective operation for heavy-duty slurry service, probably involving significant wear.

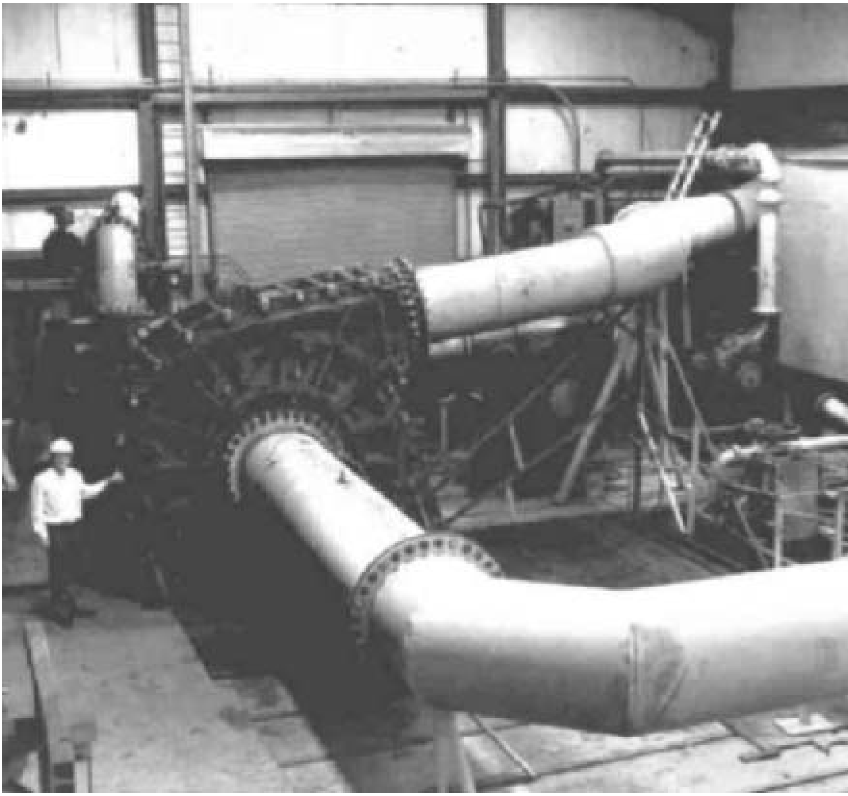

FIGURE 1 Testing a dredge pump at the GIW hydraulic laboratory

SLURRY PUMP TYPES

The majority of centrifugal slurry pumps that fit the above definition are constructed as the single-stage horizontal radially split type. A sectional arrangement drawing of a single-wall metal type is shown in Figure 3. Where elastomers are used, they must be supported with an outer casing or sideplate. This type is commonly referred to as a double-wall type. An example of a double-wall rubber-lined pump is shown as Figure 4.

Vertical cantilever and vertical submersible centrifugal pumps are also built in significant quantities and serve mostly in general cleanup and other service. These pumps on occasion are called upon to operate as transporting devices and, depending on the design and service, include the special features described later.

CENTRIFUGAL SLURRY PUMP HYDRAULIC DESIGN

In order to reduce wear, the hydraulic design of a slurry pump should be such that the rotating wetted component and fluid velocities must be kept as low as practical while hard metal or elastomer wetted-section thicknesses are increased. Impeller meridional sections are also usually kept close to rectangular and with front sealing on a vertical face to minimize the axial rotating surfaces. Collectors (casings) are usually semi-volute (or even

FIGURE 2 Dredge pump impeller, 105 in (2.67 m) diameter

annular) to give better wear over a wider range of best-efficiency-point quantity (flow rate) (BEPQ).

Where larger solids are to be pumped, the inside shroud impeller widths are usually oversized, the number of vanes is reduced to two or three, and the vane overlap may be reduced. All of these distortions modify or reduce the hydraulic performance. Given the variety of types (and severity) of services, the design solution that provides the best compromise between wear life and efficiency and provides the end user with the overall lowest total cost of ownership is not simple to achieve. Although specific to a particular slurry type, Kasztejna et al. (1986) and Cooper et al. (1987) provide a good insight into some of the design tradeoffs and special problems involved in the design of a centrifugal slurry pump.

In general, slurry pumps are of low specific speed (as defined in Section 2.1), in the range of 750 to 2000 for N_s, (14.5 to 39 for n_q and 0.27 to 0.73 for Ω_s). They employ three to five vanes, have impeller inside shroud widths that are 75–200% wider than those a normal water pump, and use impeller vane and shroud thicknesses two to three times that of a water pump. The results of these distortions flatten the constant-speed head-flow curves and reduce efficiency by five or so percent. Typical head coefficients (as defined in Section 2.1) and efficiencies are shown in Figures 5 and 6.

In order to minimize wear over different ranges of percent of BEPQ, operation slurry pump casings vary from the true volute water pump T type (shown schematically on Figure 7) through the semi-volute C type to the essentially annular A type. There are also a few examples of what is called the OB type with recessed tongue and special extended

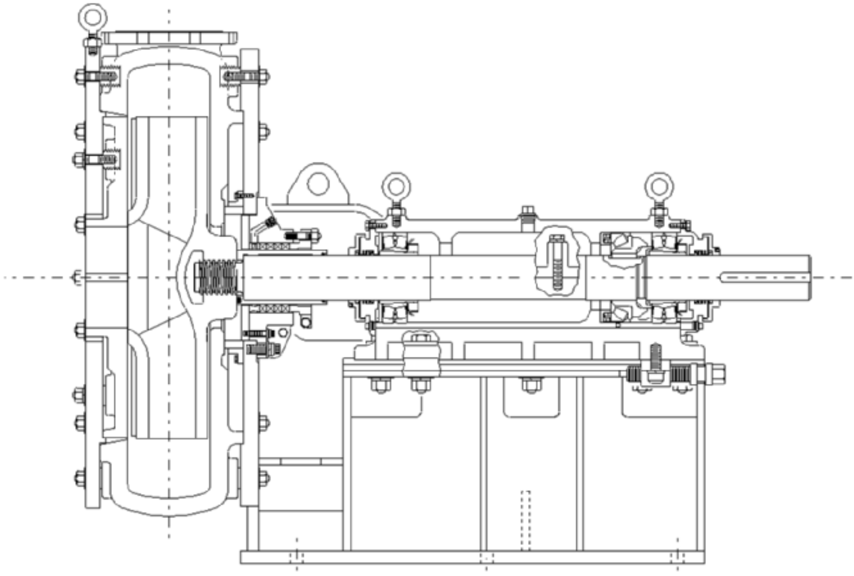

FIGURE 3 Typical single-wall hard iron slurry pump

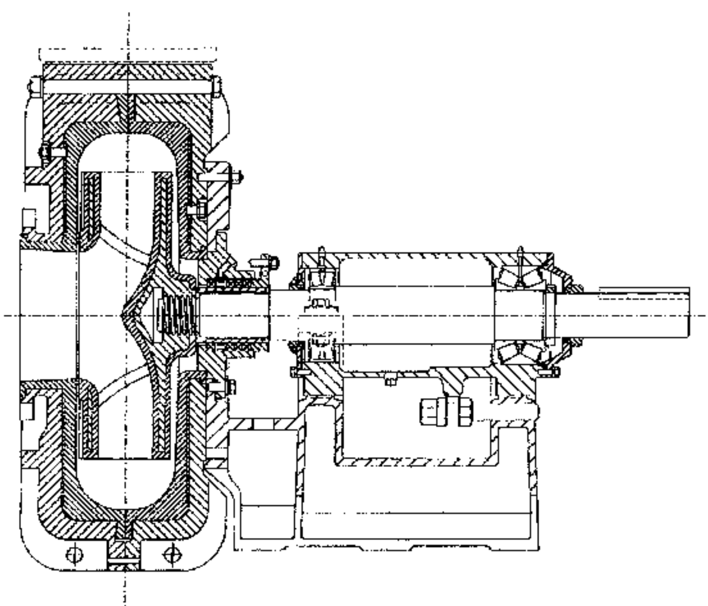

FIGURE 4 Typical double-wall elastomer-lined slurry pump

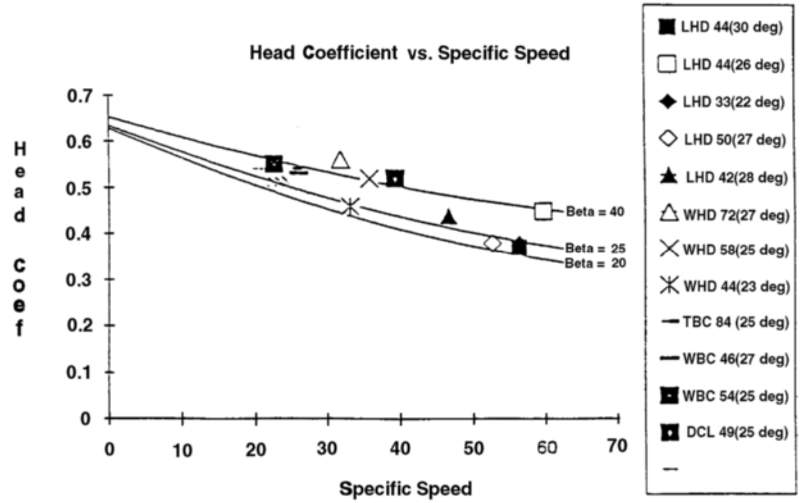

FIGURE 5 Head coefficient versus specific speed n_q, based on m³/s, m, and rpm (from Addie and Helmly, 1989). [Note: for specific speed N_s, based on gpm, ft, and rpm, multiply n_q by 51.65; for universal specific speed Ω_s, divide n_q by 52.92.]

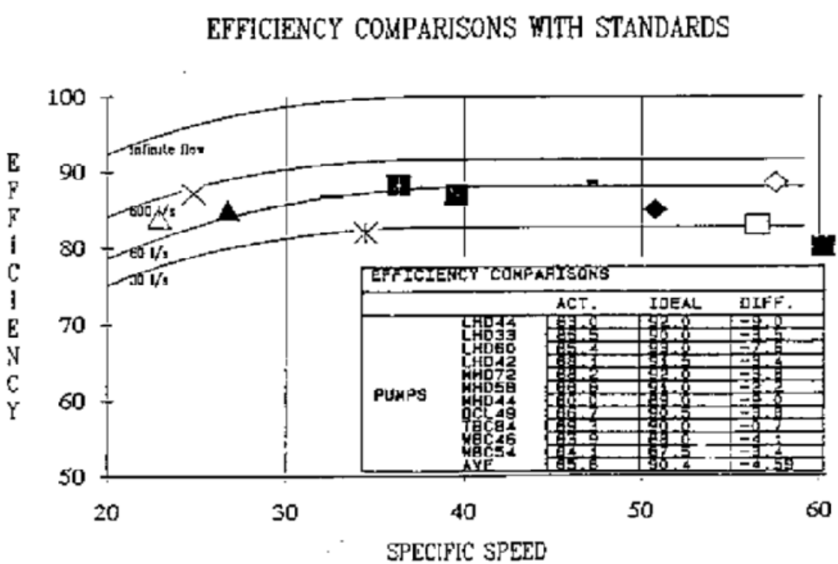

FIGURE 6 Pump efficiency as a function of specific speed n_q, based on m³/s, m, and rpm (from Addie and Helmly, 1989). [Note: for specific speed N_s, based on gpm, ft, and rpm, multiply n_q by 51.65; for universal specific speed Ω_s, divide n_q by 52.92.]

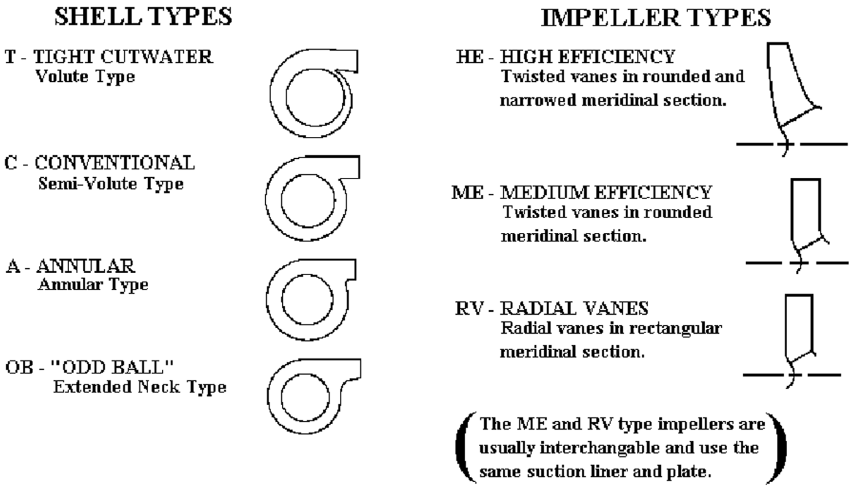

FIGURE 7 Types of shells (casings) and impellers

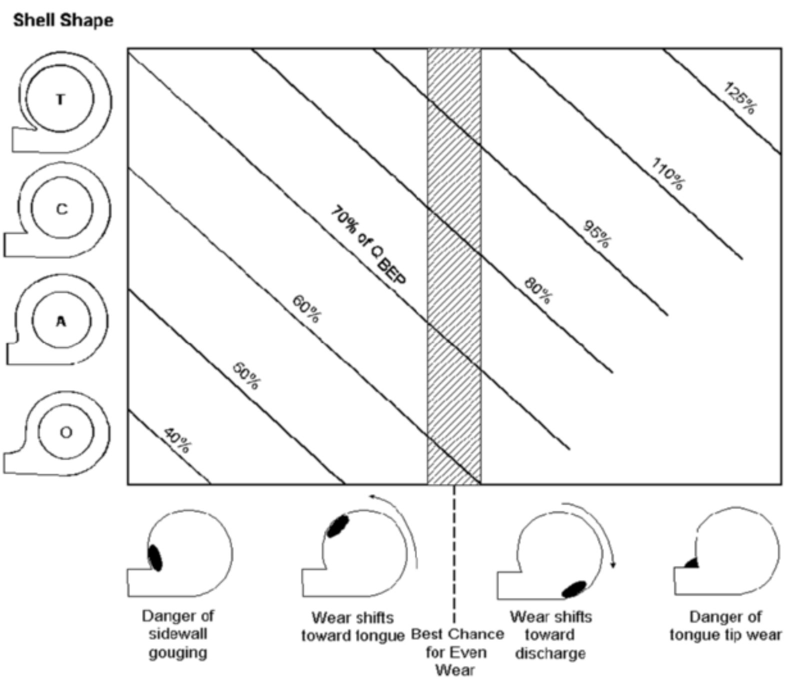

FIGURE 8 Trends in location of casing belly wear

neck intended for limiting wear in the tongue area due to excessive recirculation (which itself is symptomatic of misapplication). The general applicability of each of these is shown in Figure 8, taken from Addie and Helmly (1989).

Impeller types may be roughly categorized also as being of the old-style square meridional section radial vane RV type (still common in rubber-lined pumps where it is difficult to mold a twisted vane), the most common rounded rectangular meridional section with twisted vanes ME type, and the HE type which is closer to a water pump.

Each combination of the types illustrated in Figure 7 has its own hydraulic performance and wear characteristics. The HE/T combination generally has the highest performance, but is not necessarily the most forgiving for wear, whereas the ME/C combination is capable of respectable efficiency while at the same time having more predictable wear performance. In extremely heavy-duty wear service, operation must be at low discharge flow rates—below BEPQ. The A type shell, or even the OB type, could be best for wear where a pump has been misapplied badly. For additional information on the selection of different hydraulic types, see Addie et al. (1996).

Impeller and shells using elastomers such as rubber, neoprene, and urethane tend to be limited to impeller tip speeds less than about 75 ft/s (23 m/s), although this can rise with stiffer elastomers, at some expense to wear life. Impellers employing elastomers require higher available *NPSH* because of the thicker impeller vane sections needed, and this condition may limit their use, for example in pumping flue-gas desulphurization slurries.

MECHANICAL DESIGN OF SLURRY PUMPS

Given the distortions noted earlier the mechanical design of a slurry pump is similar to that of a water pump. Slurry specific gravities (SG) of up to 1.6 and higher may have to be pumped, so the shaft and bearings of necessity must be more robust. Figures 3 and 4 show typical pump bearing housings.

Impeller attachment by an Acme-type screw has generally been found best capable of carrying the heavy loads required, and the connections can be manufactured economically in the hard materials most commonly employed in impellers. The large loads associated with heavy-duty service require roller bearings with separate roller thrust bearings. Designs with ball bearings are suited only for light-service pumps.

The absence of radial sealing in slurry pumps allows shaft deflections larger than those found in water pumps, and these may limit the life of stuffing boxes and mechanical seals. Newer and better designs, however, tend to have shorter shafts and smaller deflections, extending the seal life. A conventional packed stuffing box is still the simplest and most common rotating-assembly seal. Configurations are similar to those used for water pumps, with the lantern-ring supplying a clear-water flush to the center for minimum dilution, or to the product side for maximum life. An expeller-type seal is popular where dilution of the product is unacceptable. These seals are limited to one stage and involve additional efficiency losses, which usually range to 3% or more.

Mechanical seals are now available that take their coolant from the product and operate with no clear-water flush; they are mostly of the single partially balanced type. In some cases, where a pump may run dry, a double type of mechanical seal must be used. At present, mechanical seals are the preferred shaft sealing method for fine-particle light-duty service such as pumps for flue gas desulphurization. These seals are also fairly widely used in the aluminum industry for red-mud pumping service. A mechanical seal described in Maciejewski *et al*. (1993) is used for pumps handling tar sands tailings at pump discharge pressures up to 350 lb/in^2 (2400 kPa), with d_{50} about 120 microns and average slurry density near 0.0578 lb/in^3 (1600kg/m^3). In this case, the process fluid is at an average temperature of about 130°F (55°C) and some 0.16 gpm (0.01 l/s) of water is used for external cooling. In this application, representative of the limits of mechanical seal technology at the present, the average seal life is about 3000 hours.

As noted before, pump casings made entirely of elastomeric materials have insufficient strength to withstand the pressure loads, so it is necessary to have an outer casing of some sort, a configuration that is commonly called a double-wall design. For versatility, some manufacturers make these designs for interchangeable wetted internal shell components of hard metal, rubber, or urethane as the service warrants. For very large slurry pumps, the double-wall configuration is heavy and costly. As a cost-effective design for this case, it is worth considering tie-bolt construction using high-tensile outer side plates over a hard-metal plate and liners as shown in Figure 1.

Further, single-wall designs with the main shell casing in hard metal are simple to construct and maintain and thus are very popular. These can be used unlined or provided with liners of bonded rubber or other material. Ceramic materials are now available that are capable of several times the life of both elastomer and hard-metal components. At the moment, the cost-effectiveness of these materials is such that their use tends to be limited to areas of high wear and other selected areas of wetted surfaces.

In a shell of single-wall metal design, the wearing components must carry the water load. Wear tends to progress until leakage of the shell occurs, but the wear is almost always localized so leakage occurs before the structural integrity of the assembly is affected. Single-wall designs may easily be ribbed or thickened locally to increase strength and wear life in a particular area. In small slurry pumps, minimum casing thicknesses usually produce safe designs of adequate pressure rating.

SELECTION OF SLURRY PUMPS

As noted earlier, service conditions vary significantly, so when selecting a slurry pump, a recommended approach is to select a service class and limit nozzle (branch) and other velocities according to that. Figures 9 and 10 from Wilson et al. (1997) outlines an approach that can be used.

For a given service class, Table 1 provides a guide to maximum impeller, nozzle (branch), and other velocities that should give reasonable wear life and keep the cost and size of the pump to a minimum.

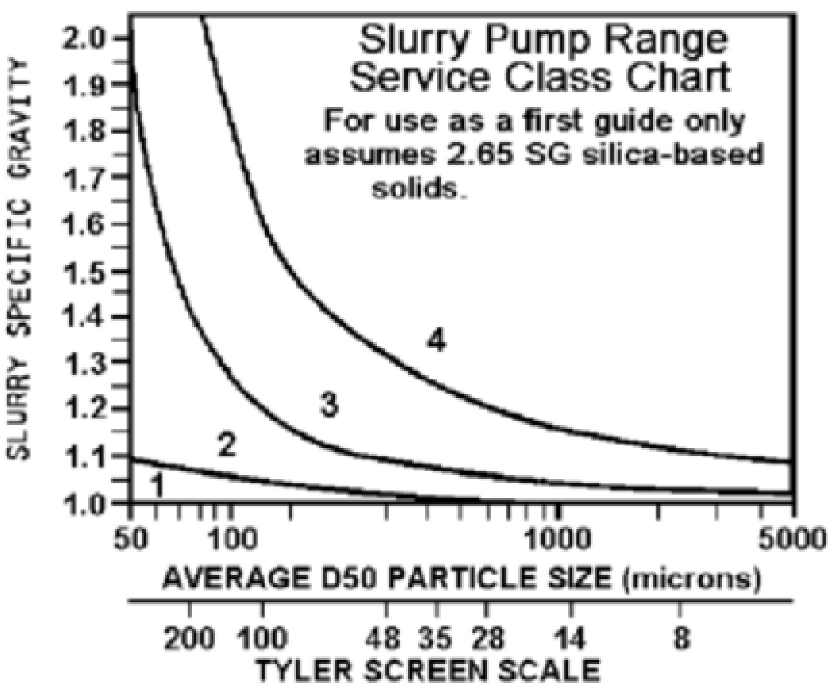

FIGURE 9 Slurry pump range service class chart

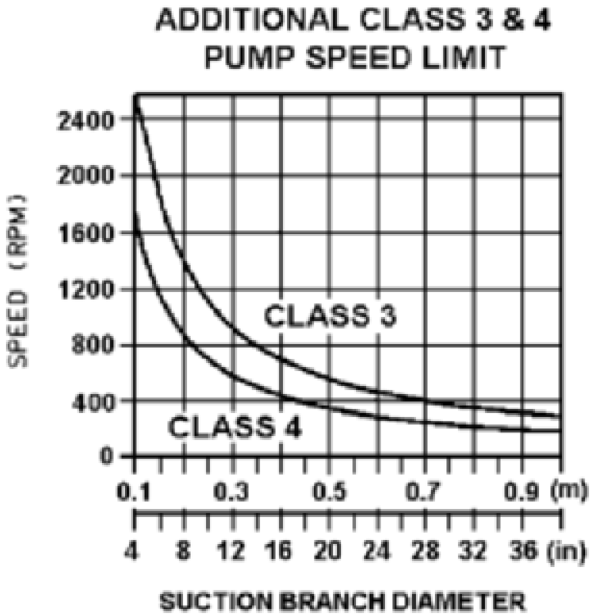

FIGURE 10 Pump speed limit chart for classes 3 and 4

TABLE 1 Recommended operating limits for slurry pumps

OPERATING* LIMITS	SHELL TYPE	SERVICE CLASS 1	2	3	4
MAXIMUM DISCHARGE (R) VELOCITY		40. (ft/s) 12.2 (m/s)	32. (ft/s) 9.8 (m/s)	27. (ft/s) 8.2 (m/s)	20. (ft/s) 6.1 (m/s)
MAXIMUM THROAT (R) VELOCITY		50. (ft/s) 15.2 (m/s)	40. (ft/s) 12.2 (m/s)	30. (ft/s) 9.1 (m/s)	20. (ft/s) 6.1 (m/s)
RECOMMENDED PERCENT RANGE (C) OF BEP FLOW RATE	ANNULAR (A)	20 - 120%	30 -110%	40 - 100%	50 - 90%
	SEMI-VOLUTE (C)	30 - 130%	40 - 120%	50 - 110%	60 - 100%
	NEAR VOLUTE (T)	50 - 140%	60 - 130%	70 - 120%	80 - 110%
	ANNULAR/OBLIQUE NECK (OB)	10 - 110%	20 - 100%	30 - 90%	40 - 80%
ALL METAL PUMP MAXIMUM IMPELLER (R) PERIPHERAL SPEED		8500 (sfpm) 43.2 (m/s)	7500 (sfpm) 38.1 (m/s)	6500 (sfpm) 33.0 (m/s)	5500 (sfpm) 27.9 (m/s)
RUBBER LINED PUMP MAXIMUM IMPELLER (R) PERIPHERAL SPEED		5500 (sfpm) 27.9 (m/s)	5000 (sfpm) 25.4 (m/s)	4500 (sfpm) 22.9 (m/s)	4000 (sfpm) 20.3 (m/s)

[Note: sfpm = ft/min; for ft/s, Divide by 60]

Except in the case of dredges (which are usually diesel driven), most slurry pumps are driven by electric motors. Because of the difficulty associated with trimming impeller diameters and variations of system head, it is common for plant slurry pumps up to 250 hp (185 kW) to have V-belt drives. For larger pumps, four pole motors and gearboxes are most common, with one or more variable speed units when there are several (usually up to six) units operating in series.

SOLIDS EFFECT ON CENTRIFUGAL PUMPS

The presence of solid particles in the flow tends to produce adverse effects on pump performance, and detailed information on these effects is needed to achieve reliable and energy-efficient operation. Head-flow curves for centrifugal pumps are often rather flat. Also, the pipeline characteristic for slurry flow usually displays a slow rise with increasing discharge. As a result, the two characteristic curves often intercept at a rather shallow angle. Therefore, even a small diminution in pump head can produce a disproportionately large drop in flow rate. The resulting difficulties in operation can cause large and expensive systems to run inefficiently or not run at all.

The effects on pump characteristics are shown schematically in Figure 11, which is a definition sketch for illustrating the reduction in head and efficiency of a centrifugal pump operating at constant rotary speed and handling a solid-water mixture. In this sketch, and the discussion that follows, η_m represents the pump efficiency in slurry service, and η_w is the clear-water equivalent. Likewise, P_m and P_w are the power requirements for slurry service and water service, respectively. The head H_m is developed in slurry service, measured in height of slurry, whereas H_w represents the head developed in water service in height of water. The head ratio H_r and the efficiency ratio η_r are defined as H_m/H_w and η_m/η_w respectively. The fractional reduction in head (the head reduction factor) is denoted by R_H and defined as $1 - H_r$; for efficiency, the fractional reduction (efficiency reduction factor) is R_η, given by $1–\eta_r$.

Test loop results for a large variety of solids at the GIW Hydraulic Testing Laboratory have been put together in a generalized design diagram (Figure 12) for larger heavy-duty pumps and smaller pumps having impeller diameters of 8 in (0.2 m) to 16 in (0.4 m). The diagram gives R_H and H_r in terms of pump impeller diameter (D) and solid average size (d_{50}) at a solids concentration by volume (C_v) of 0.15 (15%) with a solids density ratio 2.65 and a negligible amount of fine particles ($X_h = 0$). For example, a pump with an impeller diameter of 32 in (0.81 m), R_H becomes 14% ($H_r = 0.86$) for a d_{50} of 0.39 in (10 mm).

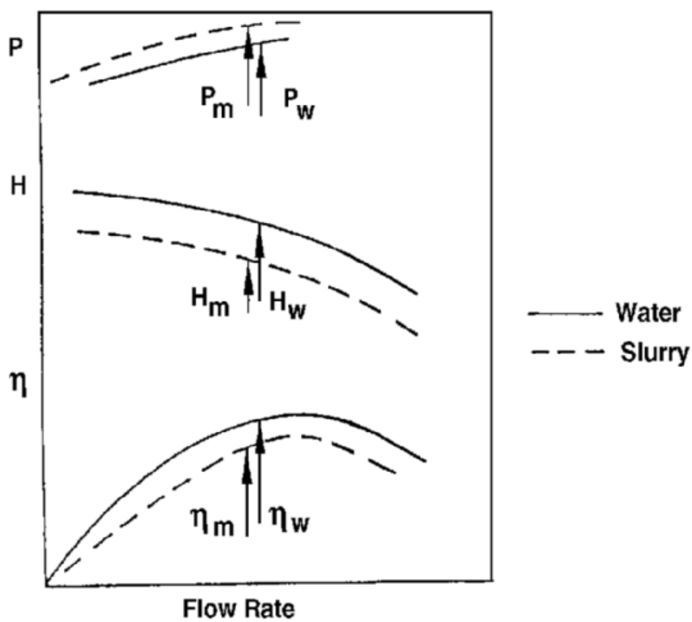

FIGURE 11 Effect of slurry on pump characteristics (schematic)

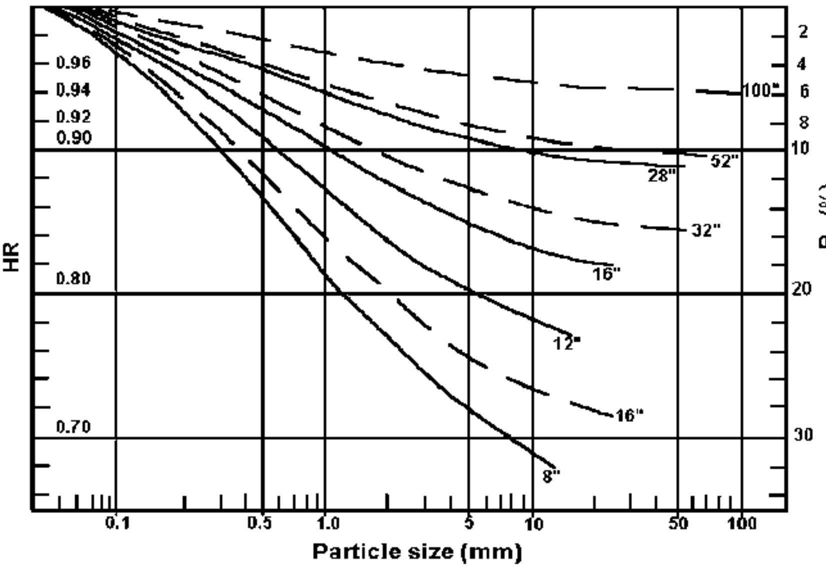

FIGURE 12 Effects of particle size (d_{50} mm) and impeller diameter (D in) on H_r and R_H. For solids concentration by volume, C = 15% with solids density ratio, S_s = 2.65 and a negligible amount of fine particles (X_b = 0). Solid lines represent smaller newer design pumps while dashed lines represent conventional heavy duty pumps. Adapted from GIW Industries Inc., U.S.A. [Note: for particle size in inches, multiply by 0.039; for impeller diameters in mm, multiply inches by 25.4.]

Corrections for various values of C_v, S_s, and X_h will now be given together with estimations of the reduction in efficiency, R_η. Based on experimental results discussed in Wilson et al. (1997), the R_H values obtained from Figure 12 may be multiplied by the following factors when the solids density ratio and the fine particle content are different from 2.65 and zero, respectively.

solids density ratio, S_s: $\left[\dfrac{S_s - 1}{1.65}\right]^{0.65}$

fine particlue content, X_h: $(1 - X_h)^2$

For example, with $S_s = 4.4$ and $X_h = 0.08$ (8%), the factors become 1.6 and 0.85, respectively. The pump previously mentioned with an impeller diameter of 36 in (0.91 m) had R_H of 14%. This value is now corrected to give

$$R_H = 14\,(1.6)\,0.85 = 19\%$$

R_H can normally be related linearly to C_v, for cases with C_v less than 15–20%. Thus with $C_v = 6\%$ in the previous example, R_H will be 19 (6/15) = 8% because the reference value for C_v in Figure 12 is 15%.

For the larger pumps in Figure 12, the reduction in efficiency, R_η, is normally less than R_H. For the smaller pumps, with $D < 16$ in (0.4 m), R_η usually equals R_H; however, it may be sensitive to solids properties. Independent of the pump type in Figure 12, R_η may exceed R_H if C_v exceeds about 20%, indicated on Figure 13.

In opposition to these tendencies, there have also been investigations where the reduction in efficiency remains small at very high concentrations, specifically with broad particle-size distributions.

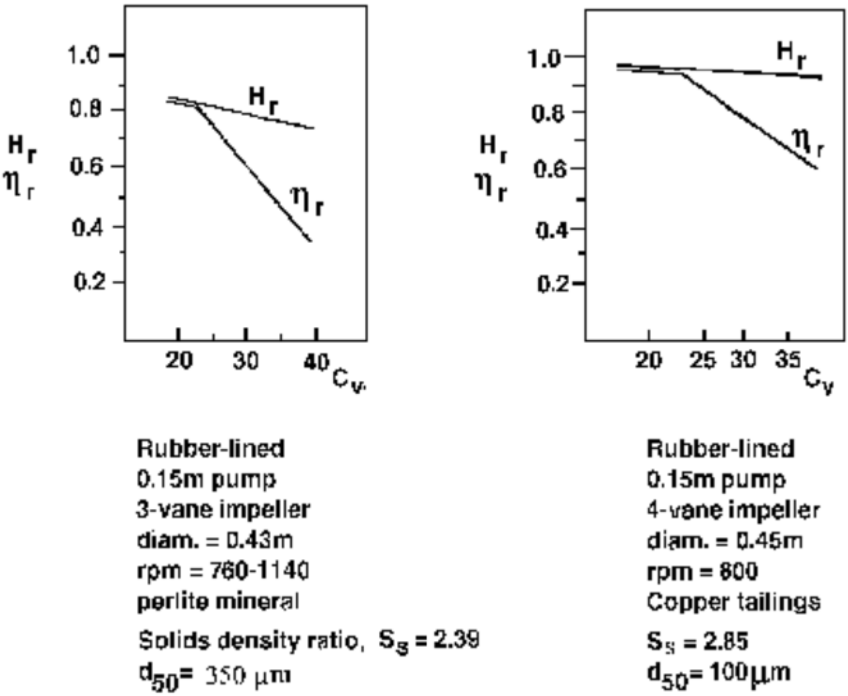

FIGURE 13 Head and efficiency reduction curves for two products (from Sellgren and Vappling, 1986). [Note: for dimensions in inches, divide mm by 25.4 and multiply m by 39.36.]

With equivalent reductions, $R_H = R_\eta$, the power consumption increases directly with the relative slurry density.

$$P_m = S_m \cdot P_w \tag{1}$$

with $R_\eta < R_H (H_r > \eta_r)$, Eq. 1 becomes

$$P_m > S_m \cdot P_w$$

that is, the power consumption becomes larger than that given by Eq. 1.

Finally, the calculated reductions may be underestimated if the solid particles are coarse and very angular. For example, the solids used in the preceding examples consisted of crushed angular particles and actually gave reductions in head and efficiency well over 10% (Sellgren et al., 1997), compared to R_H = 8% calculated here based on Figure 12.

On the other hand, with very fine particles, or large X_h-values and high solids concentrations, most slurries are practically nonsettling. In this case, the solids effect is mainly related to the rheological behavior. When pumping nonsettling slurries in industrial applications, the slurry normally behaves in a non-Newtonian way, giving widely varied pump performance effects. In general, however, small pumps are affected more than large units when pumping highly viscous or non-Newtonian media. Furthermore, the influence on the efficiency is normally larger than on the head.

Occasionally, highly non-Newtonian slurries may be pumped by centrifugal pumps operating at flow rates much lower than the maximum-efficiency value Q_{BEP}. This may cause a dramatic drop in head, which creates an unstable head curve, seen for example in Figure 14.

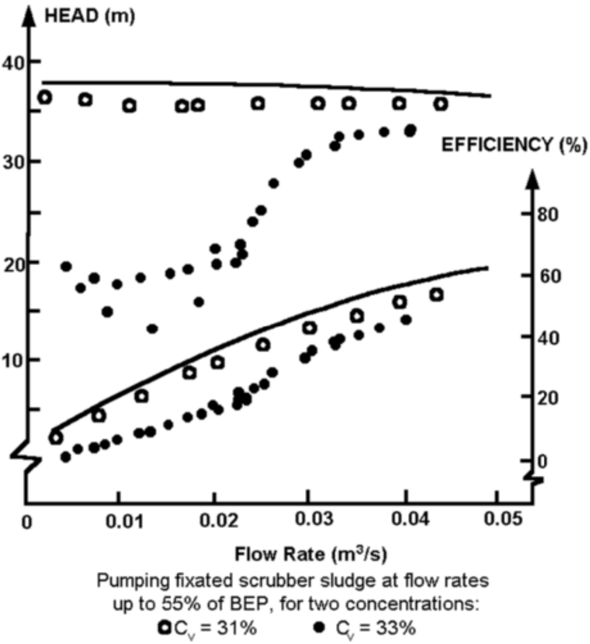

FIGURE 14 Effect of highly non-Newtonian slurries on pump head and efficiency (from Sellgren et al., 1997). [Note: for flow rate in gpm, multiply m^3/s by 15,850; for head in ft, multiply m by 3.28.]

WEAR MECHANISMS AND MATERIALS

As noted earlier, wear in a slurry pump can be severe. It is not possible, therefore, to use normal cast iron or other materials. Ideally, the life of components should be a year or more. In practice, even with the best design and materials, it can sometimes be as low as a month. Understanding wear and using the correct materials is key in the design of a slurry pump. The major erosive mechanisms are sliding abrasion and particle impact. The sliding-abrasion mode of wear typically involves a bed of contact-load particles bearing against a surface and moving tangentially to it, as illustrated on Figure 15. In pipelines,

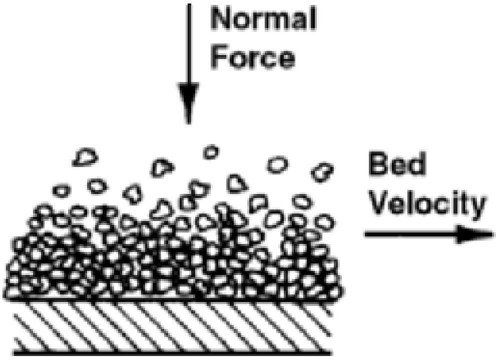

FIGURE 15 Erosion by sliding abrasion

the stress normal to the surface is caused by gravity. For sliding abrasion, the erosion rate depends on the properties of particles and wear surfaces, the normal stress, and the relative velocity.

The normal stress is enhanced when the flow streamlines are curved, as in a pump casing. In this case there is a centrifugal acceleration, equal to u^2/r, where u is the local velocity and r is the radius of curvature of the streamlines. This acceleration can often be much greater than that of gravity, producing a commensurate increase in the normal stress between the moving contact-load solids and the wall material, and hence a greatly increased rate of sliding-abrasion wear. This type of behavior can cause a pump casing to wear through when adjacent straight-pipe sections of the same material have hardly been affected by erosion. Near the tongue of the pump casing, however, the other major wear type is more significant.

The second type of wear is the particle-impact mode, which occurs where individual particles strike the wearing surface at an angle, despite the fact that the fluid component of the slurry is moving along the surface (Figure 16). Removal of material over time occurs through small-scale deformation, cutting, fatigue cracking, or a combination of these, and thus depends on the properties of both the wearing surface and the particles. Ductile materials tend to exhibit erosion primarily by deformation and cutting, with the specific type depending on the angularity of the eroding particles. Brittle or hardened materials tend to exhibit fatigue-cracking erosion under repeated particle impacts. For a given slurry, the erosion rate depends on properties of the wearing surface: hardness, ductility, toughness, and microstructure. The mean impact velocity and mean angle of impact of the solids are also important variables, as are particle characteristics such as size, shape, and hardness, and the concentration of solids near the surface.

As wear mechanisms vary considerably, so also do the engineering materials that resist them. A survey of all the available wear-resistant materials and their applications would provide subject matter for a volume in itself. Nevertheless, there are a few groups of materials that have found widespread successful application against erosion in slurry systems, and thus merit particular attention. These materials fall into the broad categories of hardened metals, elastomers (rubbers and urethanes), and ceramics. Materials in each of these

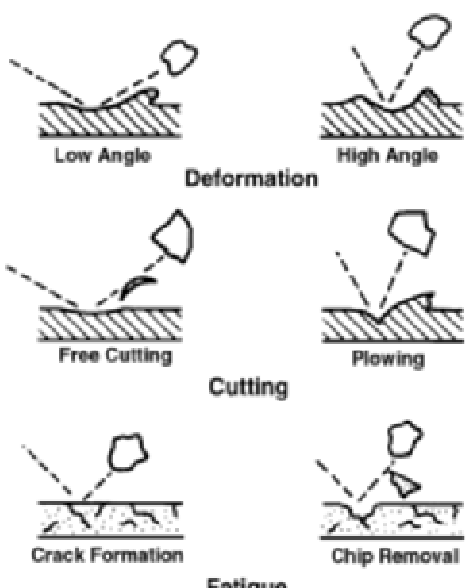

FIGURE 16 Mechanisms of particle-impact erosion

categories have their advantages and disadvantages, and the final choice of a material must account for wear resistance, strength, ease of maintenance, direct cost, and indirect costs such as plant downtime for maintenance, or failure, safety of operation, and effect on system efficiency.

Within the metals group, superior wear resistance is given by the high-alloy white cast irons, particularly nickel-chrome (for example, NiHard), chromemoly (for example, 15% chromium and 2% molybdenum), and high chrome (for example, 27% chromium) alloys. These materials exhibit higher hardness (600 plus Brinell), but less toughness than steel alloys of similar tensile strength. The high carbon content and resultant massive carbides present in these metals gives them excellent resistance to all forms of particle erosion, especially in sliding or impact at low angles (cases often encountered in slurry applications). For the coarsest slurries, hard metal is common, principally the so-called NiHard, high-chrome, or chrome-molybdenum alloys of 600–700 Brinell hardness.

For slurries without coarse particles, elastomers such as rubber, neoprene, and urethane tend to wear better than hard metal, especially for liners and shells. This improvement applies only if there is no “tramp” (that is, extraneous material such as tools, bolts, or pieces of broken castings). Elastomer selection may also be determined by the corrosive characteristics of the slurry. Natural soft rubber remains an excellent and economical pump lining material to handle fine abrasive and corrosive slurries. Carbon black can be added for additional strength, hardness, and tear resistance, to better withstand the impact of large particles in the slurry. Like all other materials, rubber has limitations that should be considered in the selection process. Wear rates in the presence of coarse material or high tip speeds may make it uneconomical, and tramp material or adverse suction conditions can tear the rubber. High temperature or the presence of oils or chemicals may require the use of one of the higher-cost synthetic materials. For example, neoprene has been found to tolerate higher temperatures and higher impeller tip speeds than can be used with natural rubber.

CALCULATION OF SLURRY PUMP WEAR

The general selection rules noted earlier aim at keeping wear to generally acceptable levels, but are otherwise non-quantitative. Finite element and other methods such as those described by Addie et al. (1996) or Pagalthivarthi (1991) are now available. They allow particle velocities inside a pump to be calculated, which, using material wear tests coefficients as described by Pagalthivarthi (1990), can be used to determine wear rates and component lives.

For example, numerical simulations allow calculation of wear around the periphery of a given pump shell for a given set of conditions. Figure 17 is an example of how wear varies with different flows from Roco et al (1983). Addie et al. (1987) shows also how a particular collector can be altered and optimized for the best possible wear.

Modeling methods of calculating wear inside a slurry pump impeller and in the nose sealing face area are also being developed so in the near future it will be possible to evaluate the wear of the complete pump in a given application. Although all of these methods cannot predict three-dimensional turbulent wear, they are now opening up the possibility of better designs, better selection, and the calculation and evaluation of the total cost of ownership, Addie et al (1998), and the benefits that could come from that.

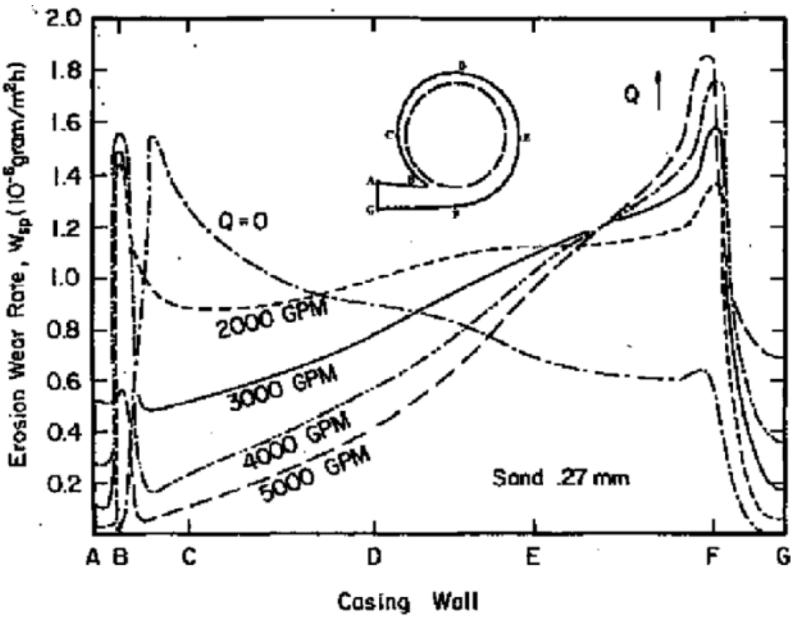

FIGURE 17 Erosion wear distributions calculated at different flow rates. [Note: for flow in l/m, multiply gpm by 3.79; for particle size in inches, divide mm by 25.4.]

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