



**FAIRBANKS NIJHUIS®**

# **SUBMERSIBLE SOLIDS HANDLING PUMPS**

## **APPLICATION AND REFERENCE DATA**

NOTE! To the installer: Please make sure you provide this manual to the owner of the equipment or to the responsible party who maintains the system.

The selection of submersible pumps for a given hydraulic requirement can vary depending on the requirements of the installation. Several pumps could meet the application criteria; however, usually a specific pump will perform best under a given set of conditions. The final selection depends on the experience, good judgement, and system knowledge by the applicator to best suit the service, system head curve, pertinent conditions, capacity range, pressure range, temperatures, and type of driver. A properly selected pump will incorporate the hydraulic and mechanical features required without unnecessary, expensive features. The reference data is from the Hydraulic Institute and will guide you to the proper pump selection.

**DEFINITIONS AND TERMINOLOGY**

**Liquid Pumped**

This should include the common identity of the liquid, pumping temperature, specific gravity, alkalinity or acidity, viscosity, vapor pressure, and solids in solution or suspension.

**Specific Gravity (S.G.)**

The ratio of the fluid's density with reference to fresh water at 39.2 F, the point at which its density is 1.0 grams per cubic centimeter.

**Capacity (Q)**

The rate of flow in U.S. gallons per minute (GPM), but can also be stated in cubic feet per second (CFS) or millions of U.S. gallons per day (MGD).

**Datum**

The horizontal plane to which all pressure measurements are referenced. The datum for a horizontal pump is the center line of the pump shaft. For a vertical pump the datum is normally the center line of the volute.

**Head (H)**

The energy level in the fluid at the location specified, referenced to the datum. The head of a pump is usually expressed in feet and is so plotted on most pump curves. Head, when expressed in feet, is the same regardless of the fluid being pumped. However, head expressed in pounds per square inch (PSI) will vary when the specific gravity of the fluids changes. The relation of feet (H) and (PSI) are as follows:

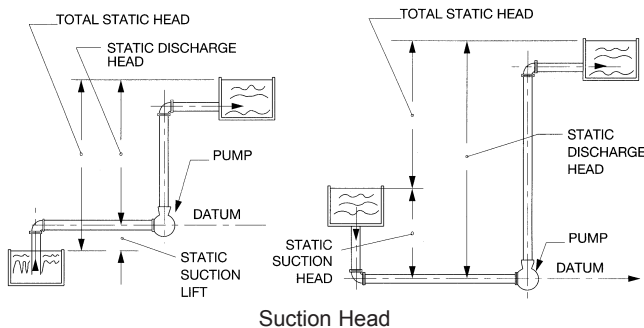
$$H = \frac{PSI \times 2.31}{S.G.} \quad PSI = \frac{H \times S.G.}{2.31}$$

**Static Suction Head**

The vertical distance in feet from the centerline of the pump (datum) to the free level of the liquid to be pumped. A positive suction head exists when the liquid is above the centerline of the pump.

**Static Discharge Head**

The vertical distance between the datum and the discharge free surface level.



**Friction Head**

The head required to overcome the resistance to flow in the pipe and fittings of the system. Major factors to consider are the liquid being pumped, the flow rate and the pipe and fitting sizes.

**Velocity Head**

A measurement of the kinetic energy of the liquid being pumped. The change in velocity head between the suction pipe and discharge pipe is considered part of the total head. For additional information regarding velocity, refer to your Hydraulic Handbook, Section I.

**Total Static Head**

The vertical distance in feet between the source of supply, free level and the discharge point at free level.

**Total Head (H)**

The total head required to be put into the pumped fluid by the pump. It includes the total static head, (±) static suction, friction head, and the fluids velocity head added.

**Pump Efficiency**

The ratio of energy delivered by the pump to the energy supplied to the pump shaft. Pump efficiency is usually expressed in percent.

$$\text{Pump Efficiency} = \frac{\text{Output}}{\text{Input}} = \frac{\text{WHP}}{\text{BHP}} = \frac{\text{GPM} \times H \times \text{S.G.}}{3960 \times \text{BHP}}$$

**Brake Horsepower (BHP)**

The actual horsepower delivered to the pump shaft.

$$\text{BHP} = \frac{\text{GPM} \times H \times \text{S.G.}}{3960 \times \text{EFF.}}$$

**Wire to Water Efficiency**

$$\text{Overall EFF.} = \text{Pump EFF.} \times \text{Motor EFF.}$$

**Absolute Pressure (PSIA)**

The pressure above absolute zero. Absolute pressure equals gauge pressure (PSIG) plus barometric pressure.

**Barometric Pressure**

Atmospheric pressure, above absolute zero, at the locality being studied.

**Gauge Pressure (PSIG)**

The pressure measured by a gauge which reads either above or below barometric pressure.

**Vacuum Pressure**

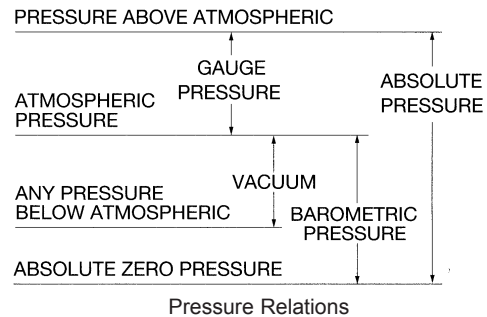
Negative gauge pressure.

**Differential Pressure**

The algebraic difference between the outlet and the inlet gauge pressures of a pump.

**Specific Speed**

Speed in revolutions per minute at which a given impeller will operate if reduced proportionately in size to deliver a capacity of one (1) GPM against a total dynamic head of (1) foot. Specific speed is used to classify impellers to their type or proportions. For additional information, refer to Section I of the Hydraulic Handbook.



### Viscosity

A measure of the internal friction tending to resist flow. It varies greatly from one liquid to another and decreases with rising temperature. For additional data on viscosity, refer to Section V of the Hydraulic Handbook.

### Rotative Speeds

Expressed in revolutions per minute (RPM), rotative speeds should be stated as a maximum limit or an operational range if a multi-speed condition is required by the application.

### Pump Mounting

Should be specified (e.g., pull-up or base mounted).

### Type of Driver

Designates the prime mover as a submersible electric motor.

### Drive Utilities and Location

Specifications include such items as electrical supply, current limitations, motor enclosure and any special location conditions. Altitude should also be specified, as power diminishes with increased altitude (only short time duty in air).

### Short Time Duty In Air

The motor can be operated exposed to air for a short time not to exceed 15 minutes (1 hour for oil filled motors) while the pump liquid is drawn down to the low level cut-off.

### Continuous Duty In Air - No Water Jacket

The motor can be operated continuously in air. Oil filled motors require a small derate; air filled motors require a larger derate. Select motors are available in oil filled construction.

### Continuous Duty In Air - Water Jacketed

The motor can be operated continuously in air. Some ratings require a small derate because of the circulator plate or external cooling system power draw.

### Inverter Applications

Brake horsepower motors used with VFD's have a 1.0 service factor.

## SELECTION INFORMATION

**Performance Curves.** Data book performance curves illustrate the expected new pump performance with normal clearances and materials of construction. The speed shown is an approximate full load motor speed. Correction due to the actual operating motor speed must be made. The listed efficiencies are those which can be expected on test when the pump is furnished with the specified amount of clear, fresh, non-aerated water not exceeding 85° F and exceeding the published net positive suction head requirements. Pump volutes or casings and impellers are finished from castings which are subject to variations in manufacturing tolerances which can affect the overall performance curve. All performance curves, if tested, meet the tolerances established by the Hydraulic Institute.

**Brake Horsepower and Driver Sizes.** Published performance curves are to be used for approximating power requirements only! Brake horsepower will be affected by the head-capacity variations allowed within the limits of the Hydraulic Institute's test code. Brake horsepower is affected also by specific gravity, viscosity, and varying voltage conditions supplied to an electric motor. All of these factors must be considered when selecting a driver and is most critical if a customer's specifications read that the driver is not to be overloaded throughout the entire curve.

**Suction Conditions.** For proper pump operation, the net positive suction head available (NPSHA) should always exceed that required by the pump (NPSHR). The type of liquid and the pump placement to the liquid source greatly influence the pump operation. Sump design is critical where adverse directional flow and insufficient submergence may create a cavitation or vortex problem.

**Pressures.** Pump design pressures have been established in accordance with sound engineering practice and are based on standard materials of construction for the pressure retaining devices such as the casing, bolting, gaskets, flanges and bearings. Any one of these items may be the deciding factor for the pump limitation.

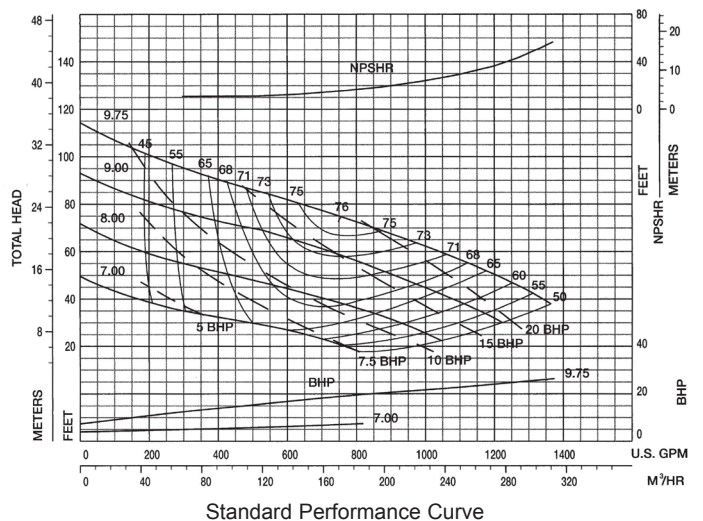
**Temperatures.** Operational temperatures have an effect on the design and construction of all centrifugal pumps. Do not exceed the ambient temperature listed on the motor nameplate.

**Materials of Construction.** The materials of construction shown in this catalog are considered standard for the environment in which the pump was designed to be operated. The various parts may be offered in materials other than that listed, however, they must be at a special quoted price and delivery.

Corrosion and erosion of pumping equipment may vary greatly depending on the application. Customers' experiences on various installations may be the best criteria for selection of materials. The factory will recommend certain materials of construction based on our experience and warranty the materials for composition and workmanship.

## PERFORMANCE CURVES

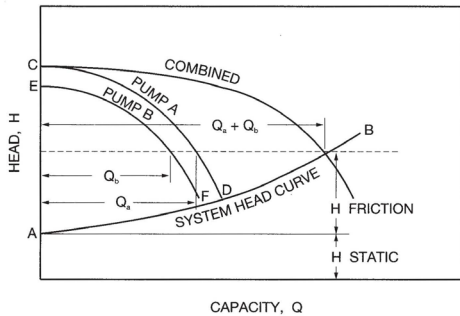
The performance of a centrifugal pump is shown graphically on a characteristic curve. Standard performance curves of pumps are plotted with total head in feet as ordinates against capacity in GPM as abscissae. A typical curve also shows brake horsepower, efficiency and net positive suction head required, all plotted over the major capacity range of the pump. Water (specific gravity of 1.0) is the liquid most often used in rating pumps. Since the head in feet developed by a centrifugal pump is independent of the specific gravity and the proposed application is figured in feet, the desired head and capacity can be read directly from the water curves without correction as long as the viscosity of the liquid is the same as that of water. The horsepower shown on the water curve will apply only to liquids with a specific gravity of 1.0. For liquids with a specific gravity other than 1.0, refer to the definition section, Brake Horsepower, for the mathematical equation to make the proper calculation.



## PARALLEL PUMP OPERATION

When the pumping requirements are variable, it may be more desirable to install several small pumps in parallel rather than a single large one. When the demand drops, one or more smaller pumps may be shut down, thus allowing the remainder to operate at or near peak efficiency. If a single large pump is used with lowered demand, the discharge valve may be throttled or the addition of variable speed and it will operate at reduced efficiency. Moreover, using smaller units provides the opportunity, during slack demand periods, to repair and maintain each pump in turn, thus avoiding plant shut-down which would be necessary with single units. Similarly, multiple pumps in series may be used when liquid must be delivered at high heads.

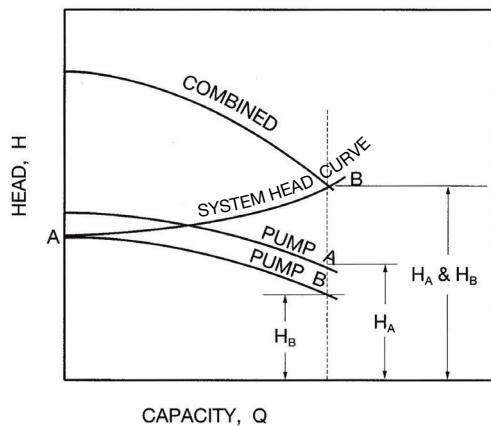
In planning such installations a head-capacity curve for the pumping system must first be drawn. The head required by the system is the sum of the static head (difference in elevation and/or its pressure equivalent) plus the variable head (friction and shock losses in the pipes, heaters, etc.). The former is usually constant for a given system whereas the latter increases approximately with the square of the flow.



Head Capacity Curves of Pumps Operating in Parallel

For units to operate satisfactorily in parallel, they must be working on the portion of the characteristic curve which drops off with increased capacity in order to secure an even flow distribution. Consider the action of two pumps operating in parallel. The system head-capacity curve AB shown above starts at H static when the flow is zero and rises parabolically with increased flow. Curve CD represents the characteristic curve of pump A; the curve for pump B is represented by EF. The combined delivery for a given head is equal to the sum of the individual capacities of the two pumps at that head. Pump B will not start delivery until the discharge pressure of pump A falls below that of the shut-off head of B (point E). For a given combined delivery head, the capacity is divided between the pumps as noted on the figures \$Q\_a\$ and \$Q\_b\$. The combined characteristic curve shown on the figure is found by plotting these summations. The combined brake horsepower curve can be found by adding the brake horsepower of pump A corresponding to \$Q\_a\$ to that of pump B corresponding to \$Q\_b\$, and plotting this at the combined flow. The efficiency curve of the combination may be determined by the following equation.

$$EFF = \frac{(Q_b \text{ GPM} + Q_a \text{ GPM}) \times H \times S.G.}{3960 \times (\text{BHP at } Q_b + \text{BHP at } Q_a)}$$



Head Capacity Curves of Pumps Operating in Series

## SUMP DESIGN-SUBMERSIBLE PUMPS

A properly conducted physical model study is the most reliable method to derive acceptable intake sump or piping designs. A physical hydraulic model study must be conducted for pump intakes when one or more of the following conditions exist:

- a non-uniform or non-symmetric approach flow to the pump sump.
- flows greater than 40,000 gpm per pump or the total station flow with all pumps running would be greater than 100,000 gpm.
- a circular station design with four or more pumps.

The model study shall be conducted by a hydraulic laboratory using personnel that have experience in modeling pump intakes.

The following information is given for reference and guidance only for preliminary sump consideration:

### Sewage Wet Well Designs

Storage capacity must consider the time liquid will be retained in the station, the optimum design of frequency of operation of the pumping equipment, and finally the storage affect of incoming lines and channels. It is most desirable to balance the pumping rates to that of the total storage capacity, keeping in mind the optimum design conditions of a pump and driver referred to a best efficiency point (BEP). The most economical and mechanically sound operation of a centrifugal pump is during continuous runs but this may result in long retention times of undesirable liquids.

In sewage stations, the shape of the wet well and the retention provided should be designed such that the total movement is maximized so that the sewage does not become septic. Most design policies base retention upon the average design rate of flow, but the maximum and minimum rates are the determining factors in sizing the wet well. In a large station requiring high capacity pumping units, the pump operation should be continuous as much as is practicable. To accomplish this, the design of the wet well must be coordinated with the sump levels at which the pumps are to start and stop. Retention time must be controlled so as to minimize the possibility of solids accumulation which could, in turn, increase the possibility and frequency of pump blockage, and to prevent solids from becoming septic, causing odors, increasing corrosion and releasing hazardous gases.

Horizontal surfaces in the wet well anywhere but directly within the influence of the pump inlets must be minimized, thereby directing all solids to a location where they may be removed by the pumping equipment. Vertical or steeply sloped sides shall be provided for transition from upstream conduits or channels to pump inlets. Trench-type wet wells and Circular Plan wet wells have been found to be suitable for this purpose.

Transitions between levels in wet wells for solids-bearing liquids shall be at steep angles (60° minimum for concrete, 45° minimum for smooth-surfaced materials such as plastic and coated concrete - all angles relative to horizontal) to prevent solids accumulations and promote movement of the material to a location within the influence of the currents entering the pump intakes. Horizontal surfaces should be eliminated where possible except near the pump inlet.

The horizontal surface immediately below the bell inlet should be limited to a small, confined space. To make cleaning more effective, the walls and floor forming the space must be confined so that currents can sweep floating and settled solids to the pump inlet.

The design must be done by an experienced wet well designer to assure that velocities are high enough to prevent the accumulation of solids while still providing the pumps with a well behaved, uniform flow to the pump inlets.

Discharge piping should not be less than the pump discharge nozzle diameter. An isolation valve should be installed in addition to a swing check type valve. Utilizing a discharge isolation valve allows inspection of the unit as well as normal maintenance work. Close attention must be given to the discharge piping and valving so as to minimize water hammer caused by the hydraulic shock waves created by quick closing discharge valves, such as a swing check type. Hydraulic shock waves

have frequently been known to magnify pump discharge pressure by as much as 7 times, occasionally 15 or more, thus placing tremendous instantaneous loads on the pumping components.

### **Circular Pump Stations (Clear Liquids)**

#### **General**

A circular pump station design is suitable for many types and sizes of pump stations. It can be used with most types of pumps including submersibles, and for most types of liquids. A circular design may offer a more compact layout that often results in reduced construction cost. Circular geometry results in a smaller circumference, and hence minimizes excavation and construction materials, for a given sump volume. Circular geometry lends itself to the use of caisson construction technique. Availability of prefabricated circular construction elements, such as concrete rings, has made this design the most popular for smaller pump stations. Fully equipped prefabricated pump stations often have circular design for the above reasons.

The recommended designs of circular stations are categorized in two groups: duplex and triplex. Stations with four or more pumps are not addressed because of their complex flow patterns. Such designs require a model study.

#### **Recommendations for Dimensioning Circular Pump Stations**

##### **Floor Clearance**

Floor clearance should not be greater than necessary, as this increases the occurrence of stagnant zones, as well as the sump depth at a given submergence. Conditions that determine the minimum floor clearance are due to the risk of increasing inlet head loss and formation of submerged vortices. Recommended floor clearance is between 0.3D and 0.5D.

##### **Wall Clearance**

Minimum clearance between an inlet bell or a pump volute and a sump wall is 0.25D or at least 4 inches (100 mm).

##### **Inlet Bell Clearance**

Minimum clearance between adjacent inlet bells or volutes (as applicable) is 0.25D or at least 4 inches (100 mm).

##### **Sump Diameter**

Minimum sump diameter should be as indicated for each pump sump figure.

##### **Volute Diameter**

This parameter is given by the proposed pump type and model. For submersible pumps with a volute in the wet pit, use the volute diameter. For end suction pumps and other pumps without a volute in the wet pit, use the inlet bell diameter.

##### **Inflow Pipe**

The inflow pipe placement minimizes air entrainment for liquid cascading down into the sump from an elevated inflow pipe. It is important to position the inflow pipe(s) radially and normal to the pumps, as shown in the figures, to minimize rotational flow patterns. There should be no valves or fittings and the inflow pipe(s) should be straight for the last five pipe diameters before entering the sump.

## **CIRCULAR PLAN WET PIT FOR SOLIDS-BEARING LIQUIDS**

#### **Wet Pit Design**

Design of the wet pit should adhere to general recommendations given in the previous section for clear liquids. Additionally, the bottom of the wet pit should have sloped surfaces around the pump inlet.

#### **Accessories**

Use of pump and sump accessories which could cause collection or entrapment of solids should be limited to an absolute minimum.

#### **Cleaning Procedure**

Frequency of cleaning cycles is dependent on local conditions, and therefore, should be determined by experience at the site. Usually, cleaning once per week is sufficient. Removal of settled solids is partially effected each time a pump is activated and brought to full speed, but removal of floating solids can only be accomplished when the water surface area is at a minimum and the pump intake submergence is low enough (0.5 to 1.0D) to create a strong surface vortex. Pumping under these severe conditions will cause noise, vibration and high loads on the impeller and hence should be limited to brief, infrequent periods. Pumps should be stopped as soon as they lose prime, or as soon as the sump is free of debris.

## **CONFINED TRENCH WET WELL DESIGN**

In this arrangement each suction inlet bell is located in a confined pocket to isolate the pump from any flow disturbances that might be generated by adjacent pumps, to restrict the area in which solids can settle, and to maintain higher velocities at the suction inlet in order to minimize the amount of solids settling out of the flow.

#### **Suction Inlet Clearance**

All suction inlets shall be located above the floor of the wet well. The side walls of the individual cell should be 1.5 to 2.0 D in dimension. The depth of the individual cell must be a minimum of 2.0 D square. A cone shall be installed under each suction inlet.

#### **Anti-Rotation Baffle**

Anti-rotation baffles are required for individual flows in excess of (3000 gpm) 189 f/s.

#### **Cleaning Procedure**

Removal of settled solids from wet wells can be achieved by operating the pumps one at a time at full speed for a duration of about two minutes. Typically, only one pump should be operated at a time to avoid excessive draw down of the liquid level in the sump.

The majority of floating solids are removed from the sump by operating the pumps one at a time at full speed while restricting the flow into the wet well to 80 to 60 percent of the capacity of the pump at full speed. Adjusting the sluice gate is the normal method of flow restriction. As the liquid level in the wet well falls, swift currents will entrain most of the floating debris, causing them to be pumped from the trench. The pump will eventually lose prime and must be stopped within ten seconds.

Both settled and floating solids are removed by the pumping equipment and discharged to the force main (or discharge conduit). This cleaning procedure momentarily subjects the pumps to vibration, dry running and other severe conditions. The frequency of cleaning cycles is dependent on local conditions, and therefore should be determined by experience at the site. Generally, the cleaning operation will take less than 5 minutes to perform and the duration between cleaning cycles would typically be 2 weeks.

## **TRENCH-TYPE WET WELLS FOR SOLIDS-BEARING LIQUIDS**

Trench-type wet wells have been successfully designed to provide for cleaning with the periodic operation of the pumping equipment using a special procedure. This standard provides guidance on the geometry necessary to induce scouring velocities during the cleaning procedure. Experience has shown that trench-type wet wells with a modified transition from the entrance conduit to the trench floor can provide optimum geometry for efficient cleaning operations. Trench-type wet wells can be used with both constant speed and variable speed pumping equipment.

#### **Inlet Transition**

The transition at the inlet to the wet well trench is an ogee spillway, designed to convert potential energy in the influent liquid to kinetic energy during the wet well cleaning cycle. The curvature at the top of the spillway should follow the trajectory of a free, horizontal jet issuing from under the sluice gate and discharging approximately 85% of the

capacity of the last pump. The radius of the curvature,  $r$ , shall be at least 2.3 times the pressure head upstream of the sluice gate during cleaning. The radius of curvature at the bottom of the ogee need only be large enough for a smooth transition to horizontal flow;  $0.5 r$  to  $1.0 r$  is sufficient.

**Inlet Floor Clearance**

All bell-type pump inlets, except that farthest from the wet well inlet, shall be located above the floor of the wet well trench. The inlet for the pump farthest from the wet well inlet shall be located above the floor of the trench.

For pumps that may be sensitive to loss of prime (due to ingestion of air from surface vortices), the pump inlet can be lowered provided the floor near the intake is lowered by the same amount.

**Inlet Splitters and Cones**

Fin-type floor splitters aligned with the axis of the trench are recommended. They must be centered under the suction bells for all but the pump inlet farthest from the wet well entrance. A floor cone should be installed under the pump inlet farthest from the wet well inlet conduit or pipe.

**Anti-Rotation Baffle**

An anti-rotation baffle at the last pump inlet is needed to ensure satisfactory performance during the cleaning cycle.

**Cleaning Procedure**

Trench-type wet wells for solids bearing liquids can be cleaned readily by stopping all pumps to store liquid for the cleaning process in the upstream conduit. When sufficient liquid is available, flow into the wet well should be limited to approximately 60 percent of the capacity of the last pump in the trench by adjusting the gate. The pumps are operated to lower the liquid level to a minimum as rapidly as possible such that the stored liquid volume is sufficient to complete the cleaning cycle. As the liquid level in the wet well falls, the liquid attains supercritical velocity as it flows down the ogee spillway, and a hydraulic jump is formed at the toe. As the hydraulic jump moves along the bottom of the trench, the jump and the swift currents entrain the settled solids, causing them to be pumped from the trench. As the hydraulic jump passes under each pump intake, the pump loses prime and should be stopped.

An alternate operation is to close the sluice gate, use the last pump to pump out the wet well, then reset the sluice gate to admit about 70% of the pump capacity when the hydraulic jump approaches the toe of the spillway.

**INLET BELL DESIGN DIAMETER**

Designing a sump to achieve favorable inflow to the pump or suction bell requires control of various sump dimensions relative to the size of the bell. For example, the clearance from the bell to the sump floor and side walls and the distance to various upstream intake features is controlled in these standards by expressing such distances in multiples of the pump or inlet bell diameter. Such standardization of conditions leading to, and around, the inlet bell reduces the probability that strong submerged vortices or excessive pre-swirl will occur. Also, the required minimum submergence to prevent strong free-surface vortices is related to the inlet bell (or pipe) diameter.

If the pump or pipe inlet has been selected prior to designing or retrofitting a sump, then the sump design process can proceed without using the information provided in this section.

If the pump (or pipe suction inlet) has not been selected, it is recommended that the inlet bell size be chosen based on achieving the average inlet velocity that experience indicates provides acceptable inflow conditions to the pump.

Permissible ranges in bell velocity in terms of the recommended bell diameter range for a given flow per pump or inlet are shown in the chart.

**'Acceptable Velocity Ranges for Inlet Bell Diameter "D"**

Pump Flow Range Q (gpm)	Recommended Inlet Design Velocity (ft/sec)	Acceptable Velocity Range (ft/sec)
< 5,000	V = 5.5	2 < V < 9
< 5,000	V = 5.5	3 < V < 8
< 20,000		
< 20,000	V = 5.5	4 < V < 7

For sump design prior to pump selection, it is recommended that the average bell diameter shown in the chart above be used. This recommended bell diameter is based on an inlet velocity of 5.5 ft./sec. (1.7 m/sec.). This process will allow the sump design to proceed. When the pump is specified and selected, the outside diameter of its bell (without added horizontal rings or umbrellas, sometimes used as vortex suppressors) should fall within the acceptable range to produce an average velocity within the limits indicated in the table above.

**REQUIRED SUBMERGENCE FOR MINIMIZING SURFACE VORTICES**

This section concerns the recommended minimum submergence of a pump bell or pipe intake to reduce the probability that strong free-surface air core vortices will occur. Submerged vortices are not believed to be related to submergence and are not considered in this section. If a submergence greater than recommended herein is needed to provide the required NPSH for the pump that greater submergence would govern and must be used.

Approach-flow skewness and the resulting circulation have a controlling influence on free-surface vortices in spite of adequate submergence. Due to the inability to predict and quantify approach flow characteristics for each particular case without resorting to hydraulic model studies, and the lack of available correlation between such characteristics and vortex strength, the recommended minimum submergence given herein is for a reasonable uniform approach flow to the pump suction bell or pipe inlet. Highly non-uniform (skewed) approach flows will require the application of vortex suppression devices. Such devices are often more practical in suppressing vortices than is increased submergence.

Even for constant flows, vortices are not steady in position or strength, usually forming and dissipating alternately. This is due to the random nature by which eddies coalesce to form coherent circulation around a filament and by which turbulence becomes sufficient to disrupt that flow pattern. For these reasons, the strength of vortices versus time should be observed to obtain an average and a maximum vortex type for given conditions.

**Controlling Parameters**

By use of dimensional analysis, it may be shown that a given vortex type, **VT**, is a function of various dimensionless parameters.

$$VT = f(F_D, N_T, S/D, I)$$

where:

- VT** = vortex type (strength and persistence)
- f** = a function
- F<sub>D</sub>** = Froude No. =  $V/(gD)^{0.5}$
- N<sub>T</sub>** = Circulation No.,  $I D/Q$ , of approach flow
- S** = Submergence
- D** = Diameter of Inlet or Bell
- I** = Circulation ( $2\pi rvt$  for concentric flow about a point with a tangential velocity **V<sub>t</sub>** at radius **r**)
- V** = Velocity at inlet (=  $4Q/\pi D^2$ )
- g** = Gravitation acceleration
- Q** = Flow

For a given geometry and approach flow pattern, the vortex strength would only vary with the remaining parameters, that is:

$$VT = f(F_D, S/D)$$

This formula indicates that a plot of **S/D** versus **F<sub>D</sub>** would contain a family of curves, each representing different values of vortex strength, **VT**. Selection of one vortex strength of concern, such as a vortex without air ingestion, would yield a unique relationship between **S/D** and **F<sub>D</sub>** which corresponds to that vortex, all for a given geometry and approach flow pattern (circulation).

For typical intake geometry and relatively uniform approach flow (i.e., low values of the circulation parameter), data and experience suggests that the following recommended relationship between submergence and the Froude number corresponds to an acceptable vortex strength.

$$S/D = 1.0 + 2.3 F_D$$

where:

- S** = Submergence above a horizontally oriented inlet plane (vertical inlet pipe) or above the centerline of a vertically oriented inlet plane (horizontal inlet pipe).
- D** = Diameter of inlet opening (equivalent diameter for nonrounded openings, giving the same area as a circular opening).
- F<sub>D</sub>** = Froude No. =  $V/(gD)^{0.5}$
- V** = Velocity at inlet face =  $Flow/ Area$

This equation indicates that one diameter of submergence must be provided, even at negligible inlet flows or velocities, and that the relative submergence, **S/D**, increases from that value as the inlet velocity increases. This is reasonable, since the inlet velocity (flow) provides the energy to potentially cause a greater vortex strength, if the relative submergence were not increased.

**Application Considerations**

For a given flow, **Q** in GPM, an inlet diameter may be selected in accordance with those parameters outlined in the previous section "Inlet Bell Design Diameter". The recommended minimum submergence for that diameter **D** would be given by:

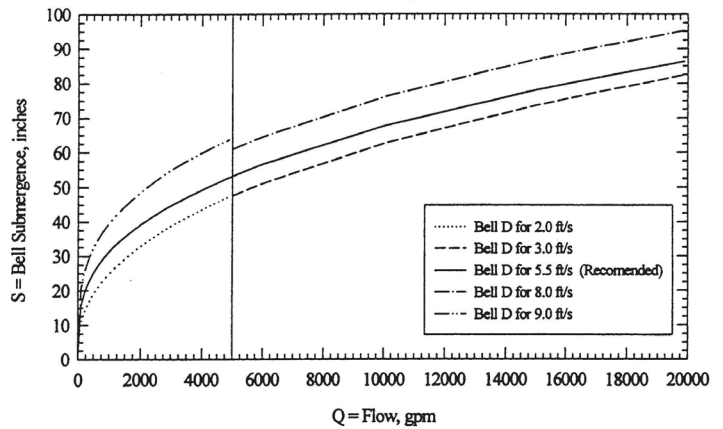
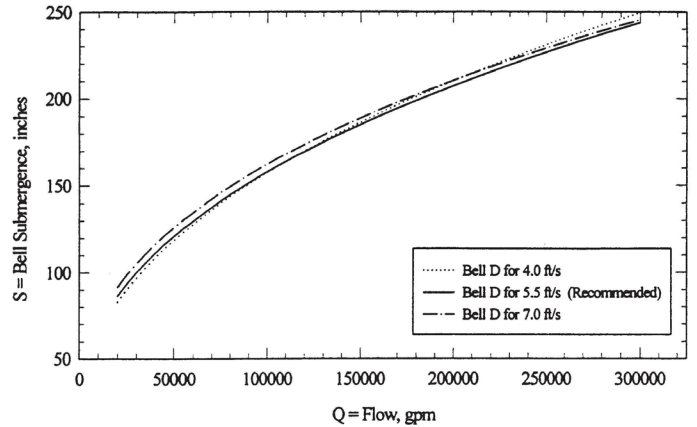
$$S = 1.0D + 2.3D(4Q/(\pi D^2)(gD)^{0.5})$$

or:

$$S = D + 0.574Q/D^{1.5}$$

using **g = 32.2 ft./sec.<sup>2</sup>** and **D** in feet. This illustrates that the actual submergence will depend on the selection of **D** for a given flow. As **D** increases, the first term causes an increase in submergence, whereas the second term causes a decrease. These opposing trends imply a minimum value of **S** with respect to **D**, allows determining that value. However, for the range of recommended bell diameters outlined in previous section "Inlet Bell Design Diameter", the required minimum submergence for reducing the severity of free-surface vortices is shown in the following graphs. This also shows the recommended minimum

submergence for the limits of the bell diameter that comply with these standards. Due to the small change in submergence, no change in submergence from that calculated with the recommended bell diameter is needed, as long as the final selected bell diameter is within the limits that comply with these standards.



**Recommended Minimum Submergence to Minimize Free Surface Vortices**

## AFFINITY LAWS

The mathematical relationship between the several variables involved in pump performance is defined by the affinity laws. These relationships which apply to all centrifugal pumps are as follows:

Law 1. With impeller diameter held constant:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \quad \text{Law 1a.}$$

$$\frac{H_1}{H_2} = \frac{(N_1)^2}{(N_2)^2} \quad \text{Law 1b.}$$

$$\frac{Bhp_1}{Bhp_2} = \frac{(N_1)^3}{(N_2)^3} \quad \text{Law 1c.}$$

Law 2. With speed held constant:

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2} \quad \text{Law 2a.}$$

$$\frac{H_1}{H_2} = \frac{(D_1)^2}{(D_2)^2} \quad \text{Law 2b.}$$

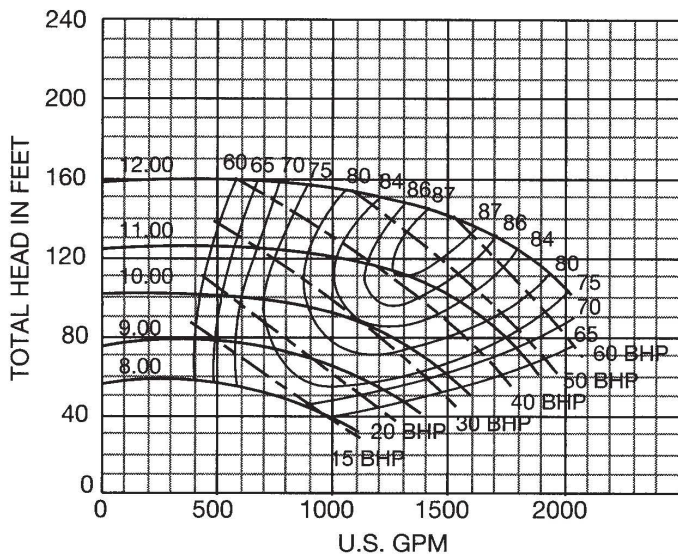
$$\frac{Bhp_1}{Bhp_2} = \frac{(D_1)^3}{(D_2)^3} \quad \text{Law 2c.}$$

Where:

- $Q_1$  = Capacity and  $H_1$  = head at  $N_1$  RPM or with impeller diameter  $D_1$ .
- $Q_2$  = Capacity and  $H_2$  = head at  $N_2$  RPM or with impeller diameter  $D_2$ .
- $Bhp_1$  = Brake horsepower at  $N_1$  or  $D_1$ .
- $Bhp_2$  = Brake horsepower at  $N_2$  or  $D_2$ .

Pump efficiency is only slightly affected by small changes in speed or diameter.

Example: The curve shows the performance of a centrifugal pump with various impeller diameters at 1750 RPM. Assume that you want to drive the pump having a 12" diameter impeller at 2000 RPM. The new pump performance can be estimated by applying the relationships under Law 1.



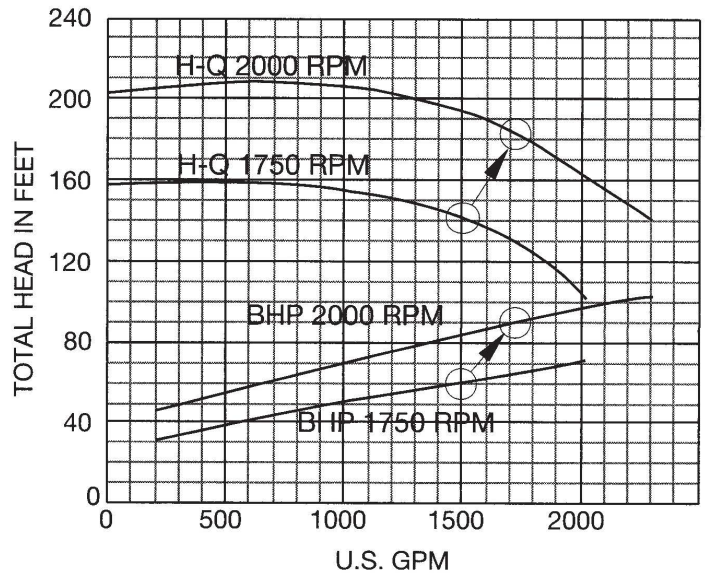
First, read  $Q$ ,  $H$ , and  $Bhp$  at several points along the 12" diameter curve in the curves below such as the best efficiency point where  $Q_1 = 1500$  GPM,  $H_1 = 140$  ft., and  $Bhp = 60$ .

Values ( $Q_2$ ,  $H_2$ , and  $Bhp_2$ ) to be used for plotting the new performance curve at 2000 RPM can then be determined by applying the relationships under Law 1. As an illustration the values ( $Q_2$ ,  $H_2$ , and  $Bhp_2$ ) at the best efficiency point of a new curve would be derived as follows:

$$\frac{1500}{Q_2} = \frac{1750}{2000} \quad Q_2 = 1714 \text{ GPM}$$

$$\frac{140}{H_2} = \frac{(1750)^2}{(2000)^2} \quad H_2 = 183 \text{ Ft.}$$

$$\frac{60}{Bhp_2} = \frac{(1750)^3}{(2000)^3} \quad Bhp_2 = 90 \text{ HP}$$



## METRIC CONVERSION FACTORS

### Head:

$$H_{(m)} = 0.102 \text{ kPa/S.G.}$$

$$H_{(ft.)} \times .3048 = \text{meters (m)}$$

### Length:

$$\text{Feet} \times .3048 = \text{Meters (m)}$$

$$\text{Inches} \times 25.4 = \text{Millimeters (mm)}$$

$$d = \text{Pipe internal diameter (mm)}$$

### Capacity:

$$\text{US GPM} \times .227 = \text{Cu. Meters/Hr.}$$

$$\text{US GPM} \times 3.785 = \text{Litres/Min.}$$

$$\text{US MGD} \times 157.73 = \text{Cu. Meters/Hr.}$$

$$l = \text{Liter}$$

$$m = \text{meter}$$

$$\text{mm} = \text{millimeters}$$

$$q = \text{Capacity (Litres/Min.)}$$

$$\text{kPa} = \text{pressure in Kilopascals}$$

$$\text{S.G.} = \text{Specific Gravity}$$

### Velocity:

$$V_{(m/s)} = 21.22 \text{ q/d}$$

### Volume:

$$\text{US Gal} \times .003785 = 1 \text{ Cu Meter}$$

$$\text{US Gal} \times 3.785 = \text{Litres}$$