C • H • A • P • T • E • R • 10

INTAKES AND SUCTION PIPING

SECTION 10.1

INTAKES, SUCTION PIPING, AND STRAINERS

WILSON L. DORNAUS
CHARLES C. HEALD

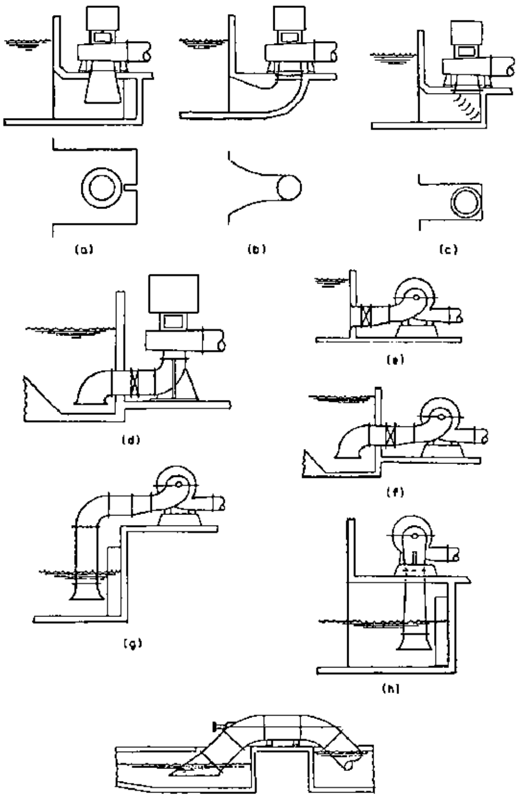
The most critical part of a system involving pumps is the suction approach, or inlet, whether in the form of piping or open pit. A centrifugal pump that lacks proper pressure or flow patterns at its inlet will not respond properly or perform to its maximum capability. Uniformity of flow and flow control to the point of pumped fluid contact with the impeller inlet vanes are the most important. Part of this may be controlled by proper pump design, but the pit designer and suction piping designer have definite responsibilities to achieve satisfactory pump operation. In open suction pit (wet-well) designs, the fluid flow must be as uniform as possible right up to contact with the pump suction bell or suction pipe, preferably without a change in direction or velocity.

Examples of dry-pit and wet-pit centrifugal pumps connected to open suction pits are shown in Figure 1.

In dry-pit pumping, the pipe leading to the pump suction flange should not include elbows close to the pump in any plane. Also, any other fittings which change flow direction and velocity and which may impart a spinning effect to the flow should be avoided within 8-10 pipe diameters of the pump suction flange. Centrifugal pumps not designed for prerotation of the fluid entering the impeller, either dry or wet pit, may suffer loss of efficiency and an increase in noise if a spinning inlet flow occurs. Fluid rotation with the direction of impeller rotation can result in a decrease in pump developed head. Fluid rotation against the direction of impeller rotation can result in an increase in pump developed head and required power, possibly overloading the driver as well as drastically affecting the pump curve shape and performance in the system. If the total system is to operate efficiently and with minimum maintenance, close attention to the suction environment of the pumps is required.

INTAKE STRUCTURES

Intake structures can be categorized as being for clear liquids or solids-bearing liquids. For clear liquids, intakes are further classified into rectangular, formed, circular, and



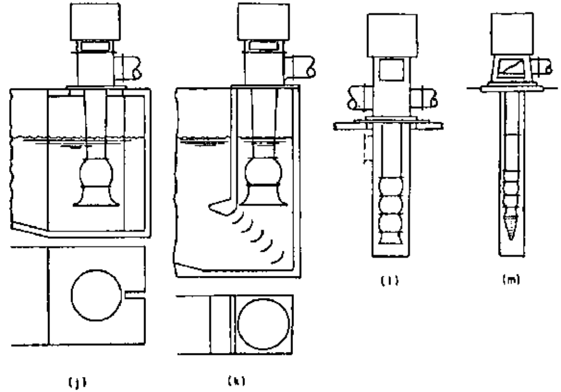


FIGURE 1 Pump suction connections: (a–d) vertical dry-pit pumps. (a) Suction bell diameter and distance from bottom, side, and back walls designed to provide radial inflow, equal velocity distribution, minimum entrance loss. Proper spacing of bell and vertical baffle required to prevent underwater eddies and rotation at inlet. (b) To minimize excavation under pump, decreasing-type suction elbow can be formed in the foundation. (c) Width of suction inlet under pump minimized by use of turning vanes. (d) Smaller pumps provided with integral suction elbows and can be connected to wet well with commercial piping. Flared suction elbow is turned downward to obtain maximum submergence at inlet. Wet well sloped to suction inlet to prevent deposition of solids.
(e–i) Horizontal dry-put pumps. (e) Pump suction connected to larger pipe with eccentric reducer (double-suction pumps only). Wet-well connection should be bell mouth of a size to provide minimum required velocity. (f) Flared suction elbow turned downward for maximum submergence over inlet. Wet-well floor may be sloped toward inlet if solids are present. (g) Vertical suction pipe with bell inlet adequately submerged to prevent vortexing. Bell sized and spaced as in Figures 1a, h, and j. (h) Vertical suction pipe connected to bottom-suction pump. Bell sized and shaped as in Figures 1a, g, and j. (i) Design used for drainage over levees. Suction pipe designed to require minimum submergency and excavation. (j–m) Vertical wet-pit pumps. (j) Distance from bottom, side, and back walls designed to provide equal velocity distribution, with splitter to prevent vortexing. (k) Width of inlet minimized by use of turning vanes. (l) Canned suction pump with either above- or below ground suction connection. Can length to suit NPSH required. Vertical installation minimizes excavation area. (m) Deep-well pump takes suction from underground water source.

trench types, as well as suction tanks and cans. For solids-bearing liquids, trench-type and rectangular wet wells are usually considered. These structures are covered in detail in American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1. Further references to this standard will be contained in this section.

COOLING WATER PUMP INTAKES

Purpose Water circulating systems must have either a continuously renewable source, such as an ocean, lake, or river, or they must recirculate the same water from cooling ponds or cooling towers. Regardless of the type of pump selected (wet or dry pit), the suction water will come from an open pit of some sort or from a pressurized pipeline.

Types of Intake

ONCE-THROUGH: OCEAN, LAKE, OR RIVER SOURCE—WET PIT Once-through intake structures are usually constructed of concrete and are arranged to gather the water into a localized area for pickup and to support the pumps. The optimum design will bring relatively clear water directly into the pump suction area at a low velocity.

It is recommended that the submergence of the pumps and the dimensions of the suction pit in the immediate vicinity of the pump suction inlet be as suggested by the pump manufacturer. Preliminary intake drawings, however, must usually be prepared for making studies and estimates and for writing specifications for equipment. During this preliminary stage of intake design for vertical wet pit pumps (or for dry-pit pumps having vertical suction pipe with a bell-mouth entrance), the recommendations found in the American National Standard for Pump Intake Design, ANSI/HI 9.9-1998 (Reference 1) should be followed. Figures 2 and 3 show the basic layout of the pumps and intake structure. Geometry is generally defined in terms of the pump inlet bell diameter, as shown. Once the number and size of pumps required is determined, a pump inlet bell diameter can be estimated. At this point, the bell diameter can be estimated based on an inlet pipe velocity of between 3 and 8 ft/s (0.9 and 2.4 m/s). The resulting pipe diameter can then be converted to a corresponding bell diameter approximately 1.5–2.0 times the inside pipe diameter. With the bell diameter selected, the proportions of the inlet structure can be estimated from Figures 2 and 3. Table 1 gives recommended values for the dimensions. For establishing velocities, the minimum submergence over the suction bell for vertical wet-pit pumps can be estimated from Figure 4, based on maximum expected flow rate.

Once the intake pump manufacturer has been selected, final intake dimensions can be established on the basis of actual pump inlet pipe dimensions and the equipment supplier's recommendations. Should there be an appreciable variation in any of the dimensions from various sources, a model test of the intake structure is justified and recommended.

For further guidance, ANSI/HI 9.8-1998 (Reference 1) provides intake design recommendations for both suction pipes and all types of wet pits that are a result of the combined efforts of sump designers, hydraulic researchers, pump manufacturers, and end users. It is intended to provide designers, owners, and users of pumping facilities a foundation upon which to develop functional and economical pumping facility designs.

The intake design process is intended to arrive at a cross-sectional area such that straight-line flow to the bell area is at an average velocity of 1 ft/s (0.3 m/s) or less. This assumes the source to be either a lake or a river with a maximum velocity of 2 ft/s (0.6 m/s). For higher velocities, correspondingly greater distances should be used to the trash rack or screen. The choice of providing a trash rack or screen, or both, is based on the type and amount of debris likely to be encountered at the inlet.

An ocean inlet has additional requirements because of tidal action and variable direction currents which may exist. For these situations, it may be necessary to create a forebay inlet basin. Such a basin would be independent of the pump pit and fed by a submerged inlet tube extending out into the ocean for some distance (Figure 5). This tube may utilize normal pipe velocities, but the inlet must turn upward and the opening should be protected by a horizontal cap to allow the water to travel horizontally at a velocity high enough to scare

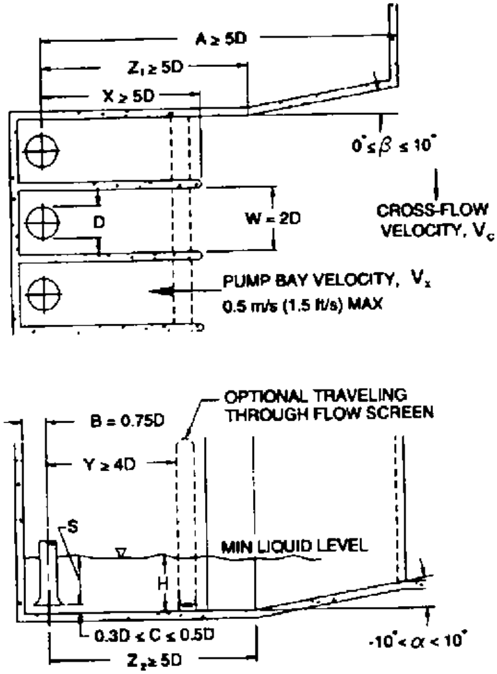


FIGURE 2 Recommended intake structure layout (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

fish away. The discharge into the forebay can be pipe velocity because the turbulence will be dissipated by lower outlet velocities [2 ft/s (0.6 m/s)] through trash racks and traveling screens. From this point on to the pump chamber, the velocity should be low and constant.

Any necessary change in inlet channel dimensions should be made gradually. Tapering walls should not diverge at more than a 14 degree included angle. If it is necessary to slope the floor, a maximum of 7 degrees is recommended and, if possible, the floor should level off before reaching the pump area, as far back as possible. No sharp drops (waterfall effect) should be permitted.

If the inlet channel is a closed pipe with full-wetted perimeter, pipe velocities can be used up to a distance from the pump chamber where the tapered wall rule can be applied. This means that the pipe size would be increased in accordance with the rules for maximum taper rate to ensure that the velocity as it discharges into the pump chamber is not over 1.5 ft/s (0.46 m/s). Unless the inlet velocity into the pump chamber is kept below 1.5 ft/s (0.46 m/s), extensive baffling and additional space will be necessary, and this can only be effectively determined by model testing of the pit.

RECIRCULATING SYSTEM—COOLING TOWER For cooling tower systems, the pump pit is normally adjacent to the tower basin. Cooling tower basins are shallow, usually not more than 6 ft (1.8 m) water depth. Because pumps will require more submergence than this, a sloping ramp or sharp drop-off will be required if pumps are to be placed close to the basin. This will save initial costs for excavation and concrete work, but the resulting poor flow patterns conditions will cause endless pump problems.

Some successful small installations have been built with downslopes of 15 degrees or more, but tower basin outlet velocity has been no more than 2 ft/s (0.6 m/s) and the downslope widened as it approaches the pump pit. For low flows, low velocities, and fortunate

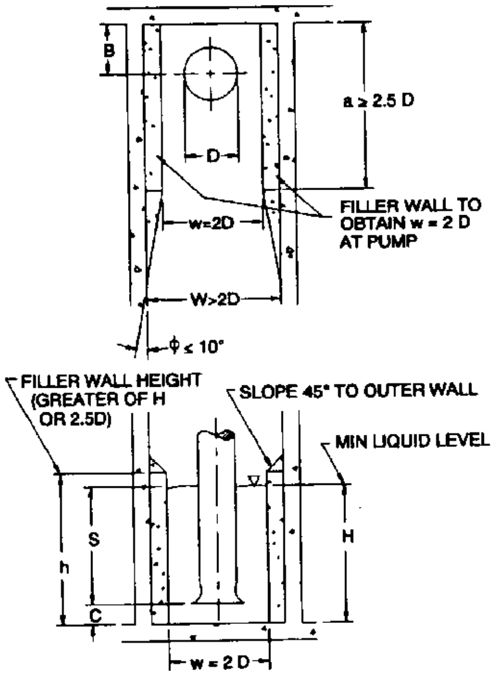


FIGURE 3 Filler wall details for proper bay width (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

arrangements of tower piers (lack of recirculation piping, and so on), such installations may be operated with reasonable success. Others of similar design, however, have proven to have disastrous effects on pump performance. Impellers and bearings may suffer rapid deterioration under such conditions.

The best solution, but obviously the most expensive solution, is to build the pump pit far enough from the tower basin to obtain a flat channel bottom. The channel should have an average channel velocity of 1 ft/s (0.3 m/s) or less for a distance at least equal to A in Figure 2, whether trash racks and screens are available. The slope from the basin floor to the channel bottom should not be more than 10 degrees, preferably less than 7 degrees. The basin exit width should be such as to make the exit velocity less than 2.5 ft/s (0.75 m/s). If several tower piers are in the exit path, this velocity limit should be reduced to 2 ft/s (0.6 m/s) or less. Upstream sides of the piers should be rounded off and the downstream sides should be tapered. Make the sidewalls of the downslope diverging so the velocity at the bottom is no more than 1 ft/s (0.3 m/s).

Velocities should be based on the runout flow of one pump. All dimensions and velocities mentioned are limits. If they are used based on maximum required flow rate, and the pump is selected for reasonable velocities, operation should be satisfactory. Obviously, if a pump with high velocities (suction bell inlet, impeller eye, rotative speed) is selected, it will require more submergence than a more conservatively designed machine. This will increase the cost by requiring a deeper pump pit located further away from the tower basin in order to perform satisfactorily.

A model test of the inlet structure is highly recommended if the best solution seems too expensive, or if topography will not allow the ideal arrangement. A model with a variable slope, variable sidewalls, and true representation of piers, pipes, and other obstructions should enable a satisfactory design to be achieved with a possible net reduction in overall

TABLE 1 Recommended dimensions for Figures 2 and 3 (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

Dimension Variable	Description	Recommended Value
A	Distance from the pump inlet bell centerline to the intake structure entrance	A = 5D minimum, assuming no significant cross-flow[a] at the entrance of the intake structure
a	Length of constricted bay section near the pump inlet	a = 2.5D minimum
B	Distance from the back wall to the pump inlet bell centerline	B = 0.75D
C	Distance between the inlet bell and floor	C = 0.3D to 0.5 D
D	Inlet bell design outside diameter	(see text)
H	Minimum liquid depth	H = S + C
h	Minimum height of constricted bay section near the pump inlet bell	h = (greater of H or 2.5D)
S	Minimum pump inlet bell submergence	$S = D\,(1.0 + 2.3\,F_D)$
W	Pump inlet bay entrance width	W = 2D minimum
w	Constricted bay width near the pump inlet bell	w = 2D
X	Pump inlet bay length	X = 5D minimum, assuming no significant cross-flow at the entrance to the intake structure
Y	Distance from pump inlet bell centerline to the through-flow traveling screen	Y = 4D minimum. Dual-flow screens require a model study
Z_1	Distance from pump inlet bell centerline to diverging walls	Z_1 = 6D minimum, assuming no significant cross-flow[a] at the entrance to the intake structure
Z_2	Distance from inlet bell centerline to sloping floor	Z_2 = 5D minimum
α	Angle of floor scope	$\alpha = -10$ to $+10$ degrees
β	Angle of wall convergence	$\beta = 0$ to $+10$ degrees (Negative values of β, if used, require flow distribution devices developed through a physical model study)
ϕ	Angle of convergence from constricted area to bay walls	ϕ = 10 degree maximum

[a]Cross-flow is considered significant when $V_C > 0.5\ V_X$ average

cost. Actual pumps and valves should be used to create the flow in the pump pit, and extremes of flow and submergence should be included in the test program.

Sometimes the cooling towers are on a hill, so a substantial drop in elevation provides pressure available nearer the flow area. If dry-pit pumps are used, this pressure can be utilized to reduce pump requirements. When this is done, the pump suction becomes a pipe, and the latter part of this section should be consulted.

RECIRCULATING SYSTEM—POND WET PIT The intake structure in a cooling pond should be located as far as possible from the inlet pipe to the pond to generate the maximum cooling effect. If spray surface equipment is used, it should be so arranged in relation to the intake building that a minimum surface disturbance is encountered. Prevailing winds should be considered, and the building should be located on the lee (upwind) side of the

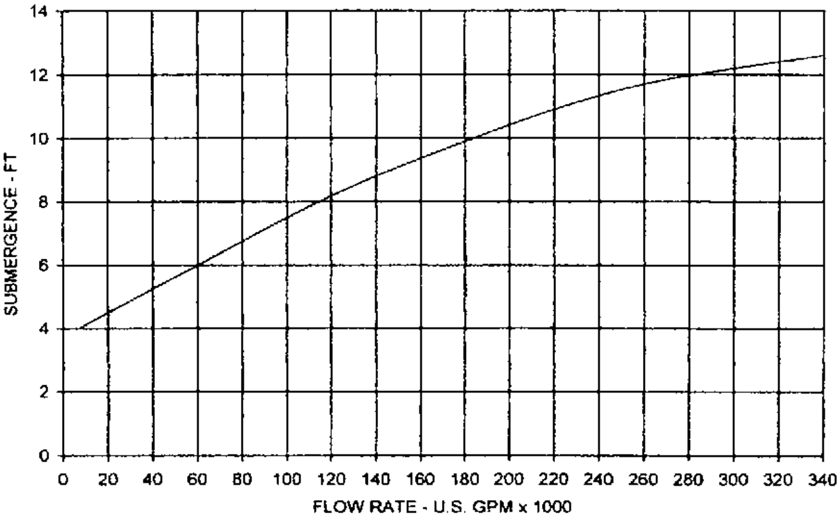


FIGURE 4 Flow rate vs. submergence over suction bell for vertical wet pit pumps. (U.S. gpm × 0.227 = m^3/h, ft × 0.305 = m).

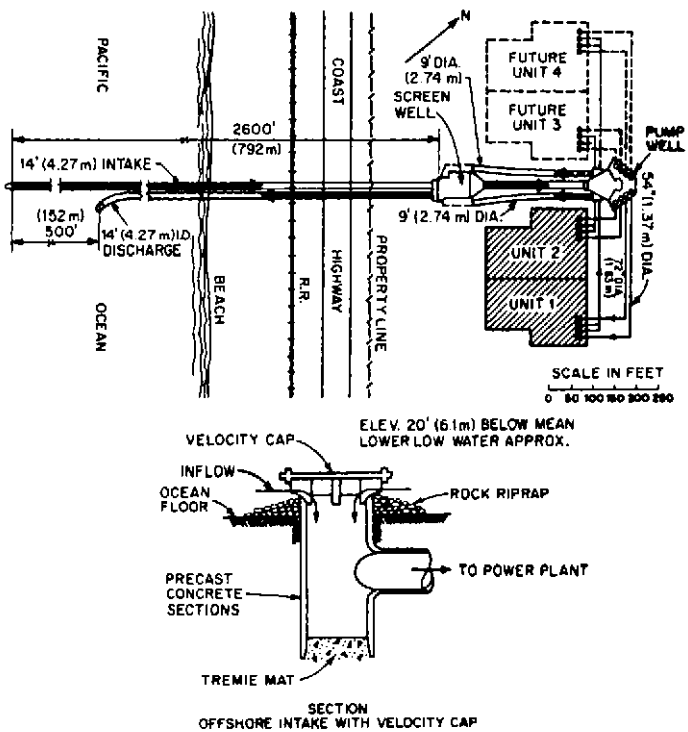


FIGURE 5 Ocean intake with velocity cap to minimize fish pickup

pond. If side and bottom areas might be easily disturbed to include silt in the flow, riprap—a wall or covering of stones thrown together randomly—should be applied to approach slopes as well as to bottom mats well beyond the inlet wing walls.

The handling of silt is usually not desirable in a pumping system. A high velocity through the pump will accelerate wear. At low velocity points in the system, silt will settle out and produce higher velocities, and more wear will occur as a result of the area blockage. If space is available, a silt-settling basin can be constructed ahead of the inlet basin. Cross baffles should be provided to slow the inlet flow to a velocity less than 1 ft/s (0.3 m/s). Most stream debris will settle out at this velocity, and the flow into the pump suction pit will be relatively clean, preventing the deposit of additional silt around the pump suction bell. If space is not available for silt beds in large-capacity installations, the main channel can be furnished with a weir across the flow path. The height of the weir should be selected to give an overflow depth above the weir of not more than one-third to one-fourth the water depth just preceding the weir. The velocity over the crest should not exceed the intake channel velocity. Weirs of this type are particularly effective when the intake channel is at right angles to the supply mainstream.

In areas where river levels vary considerably throughout the year, problems arise not only from silt and debris accumulation, but from the structures required to prevent damage to motors and electrical switch gear. If the required flow is moderate, a system known as the Ranney well (Figure 6) can be constructed. A Ranney well is a concrete silo 13 ft (4 m) in diameter, which becomes the collecting basin and pump well. It may be situated in or near a river and can be partly below and partly above water level with settings up to 100 ft (30 m). Small perforated pipes radiate horizontally from the base of the well and tap

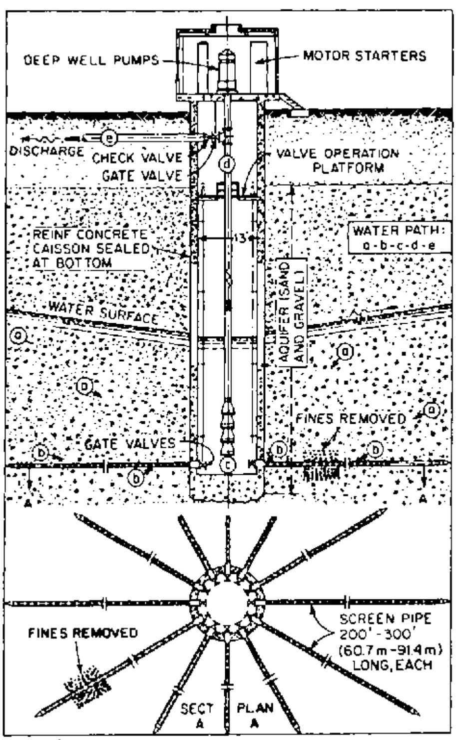


FIGURE 6 Ranney well collects into a pool from underground strata (Ranney Method Division, Pentron Industries)

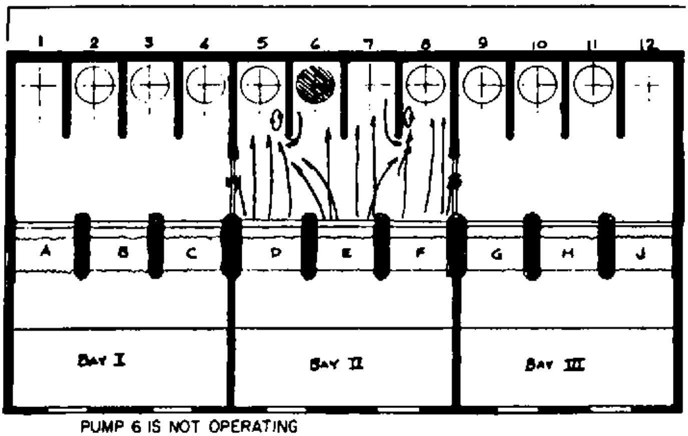


FIGURE 7 Empty spaces on nonoperating pumps cause vortexing at wall ends because of flow reversal.

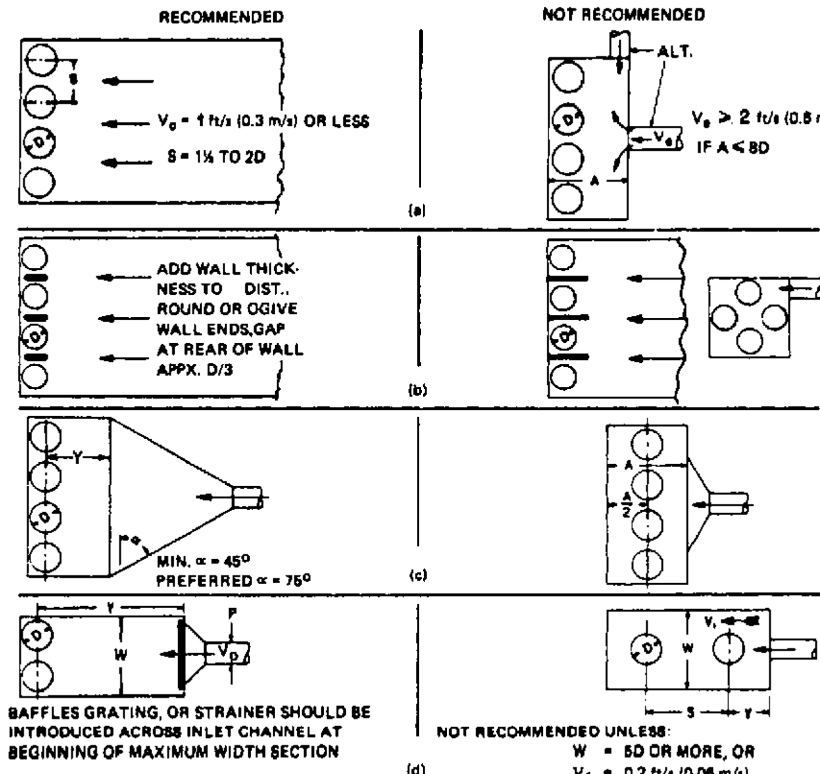
into porous strata, bringing small flows of water into the main well. This water can then be pumped from the well with a deep-well wet-pit vertical pump.

Multiple Pumps Some pump requirements are easily met with 100% capacity single pumps, but more reliability usually requires two 50% or three 33% pumps, or, if the service is sufficiently critical, three 50% or four 33% pumps, and so on. There are practical limits of size for various pump types, so large flow demands will undoubtedly call for a multiple pump arrangement.

The problem arising from multiple pumps arranged in a common pit is from the probable nonuse of some of the pumps while others are operating. This can cause variables in flow patterns that may lead to eddying and vortexing (Figure 7). Installation of separating walls in the common pit may introduce additional problems because the ends of the separating walls can create eddy currents in the corners at the unused pumps. A back vent in the dividing wall will relieve this situation, provided it vents at the water surface (Figure 8b left). If walls are extended past the screens and trash racks to a forebay, this problem will not occur, but the design has then become that of a single-pump basin.

The same velocity rules apply to multiple arrangements as to single-pump basins. Odd arrangements should be avoided even when they look invitingly symmetric—fan-shaped, round, radial, peripheral—all have directional problems that are not easily overcome. A basic pit design, consisting of a number of equally-sized pumps in a common pit basin with flow entering parallel and straight in at 1 ft/s (0.3 m/s) or less, would not need to be model tested to assure reliability (Figure 8a left). If separating walls are required for structural support, and they are properly shaped and vented, no model test is recommended. The pumps should be located at the extreme rear of the pit so the whole approach assumes the characteristics of a suction pipe. Individual pump manufacturers may vary the location of the pump relative to the pit bottom, velocity of inlet spacing, and so on. It has been found that some of these variations require additional splitters or baffles below the pump, up the back wall behind the pump, or centered in the flow ahead of the pump. If so, a model test should be run and the additional pit cost weighed against other alternatives (changing the pit shape, pump location, pump size, and pump speed).

A dry-pit pump installation will have the pumps located either in a dry well at or below wet well water level (Figures 1e and 1f) or directly above the wet well and using a suction lift (Figures 1g and 1h), which calls for priming equipment. The additional cost of priming equipment (vacuum pumps, and so on) may be partially offset by the additional space and valve requirement of the first option. In either case, the suction piping in the wet well



W/P	1.0	1.5	2.5	4.0	10.0
Y	3D	5D	8D	10D	15D
V_P	1	2	4	6	8

FIGURE 8 Using basics of good pit design precludes the need for model testing (Hydraulic Institute Standards, 13th Edition, 1975—out of print).

should be treated in the same fashion as the pump suction bells in wet-pit installations as far as spacing, direction, and velocity of flow are concerned.

When the pump is installed at an elevation that may be below the water level in the suction pit, a valve must be installed at the suction of the pump. The temptation to reduce the inlet size to use a smaller valve should be avoided. Pipe size at a pump inlet may decrease into the pump down to the pump suction size, but it would not be reduced below that size and then have to increase again as it enters the pump. The pipe in the wet well should preferably have a bell end and project downward. The minimum water level above the top edge of the pipe or the lip of the bell should be at least 5 ft (1.5 m) for a recommended entrance velocity of 5 ft/s (1.5 m/s). The bell mouth should protect downward to assure uniform inlet flow and to attain maximum submergence.

A wet-pit intake style that closely approximates a suction pipe arrangement and uses what is essentially a dry-pit pump has been expanded by the U.S. Department of the Interior, Bureau of Reclamation to an elbow-type suction tube design. This incorporates a formed concrete suction inlet with a swing of 135 degrees in the vertical plane and a gradual decrease in area to the suction eye of the pump (Figure 9). The resulting design saves considerable excavation in the wet well area, reduces losses in the pump,

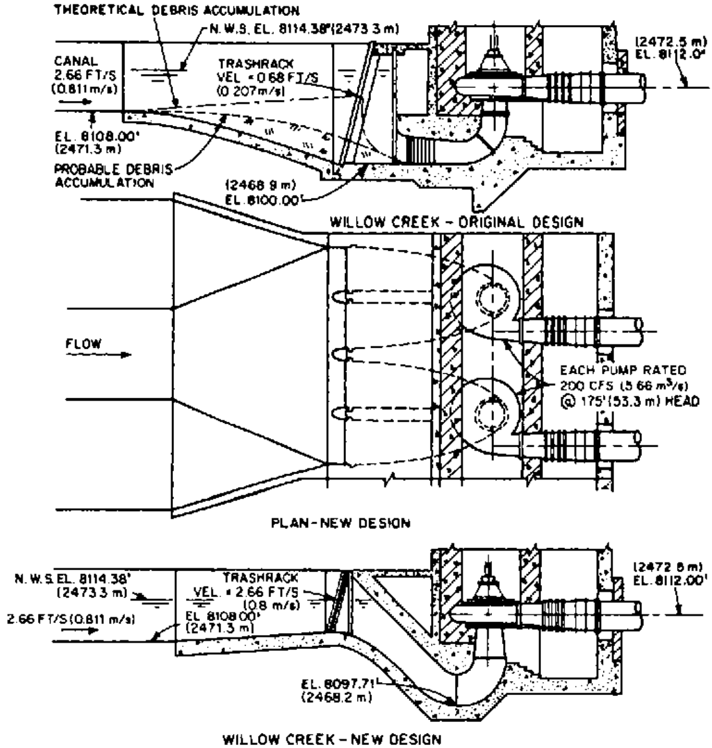


FIGURE 9 Improved 135° design of Bureau of Reclamation elbow suction tube inlet (U.S. Department of the Interior, Bureau of Reclamation)

and allows a smaller, higher-speed pump to be used. With higher velocities, debris dropout is reduced so silt buildup does not occur as readily and a smaller trashrack area remains effective longer. These inlets are independently self-sufficient and may be grouped into multiples as long as the forebay is designed to adhere to the basic design rules for wet-pit approach channels. Additional guidelines on the proportions of formed suction intakes are included in the Hydraulic Institute Pump Intake Design Standard.

Considerations for either single- or multiple-pump pit design relate primarily to even flow and low velocity into the pump. At the leading edge of the pump impeller (suction) vanes, this velocity will be increased and the direction of flow violently changed. To make the transition from pit flow to pump flow is the work of the pump designer. Some pump designs include ribs in the suction bell; others do not. It is obvious that the transition from parallel flow at 1 ft/s (0.3 m/s) to right-angle rotating flow at 15 to 20 ft/s (4.6 to 6.1 m/s) requires a high degree of skill in matching the suction bell to the impeller. If the pump is not designed to handle a wet-pit installation as described in previous paragraphs, a turning vane pit may be required.

Suction Pit Turning Vanes Figure 1c illustrates how turning vanes are used to guide the flow of water into the inlet of a vertical volute dry-pit pump. Figure 1k shows the same for the suction bell of a vertical diffuser wet-pit pump. If these pumps were connected to an inlet sump without turning vanes, as illustrated in Figures 1a and 1j, a wider and longer channel would be required to feed the suction bell from all directions. The pumps with turning vanes an narrower inlet channels may be desirable as multiple pumps can be spaced closer together providing screen width requirements do not dictate spacing.

Comparing depth of pit bottom below pumps with and without turning vanes, the following is to be noted. The excavation beneath a vertical volute dry-pit pump with turning vanes can be slightly less than the excavation beneath the same pump with no vanes but with a suction bell. The velocity approach into the closed portion of the channel beneath the pump can be as high as 3 ft/s (0.9 m/s) if turning vanes are used at the design flow; but it should be limited to 1.5 ft/s (0.46 m/s) with no turning vanes. Although the suction bell design requires a wider channel than a vaned inlet (approximately two bell diameters), this is not wide enough and the channel must be made deeper to meet the lower velocity requirement for this type of inlet. When turning vanes are used with a vertical diffuser wet-pit pump, the pit must be excavated deeper than would be required if no vanes were used. This additional depth is required to form an elbow in the narrower channel and provide equal flow distribution to the impeller. Design velocity at the inlet vanes is 3 ft/s (0.9 m/s).

The setting of the lip of the suction bell and the pump impeller below design low-water level for volute dry-pit pumps must be the greater of the dimensions required to

- Prevent vortexing (ANSI/HI dimensions)
- Provide adequate *NPSH* at the centerline of the impeller
- Provide a level of water sufficient for the unit (impeller) to be self-priming

As an alternative to the use of turning vanes under a vertical volute dry-pit pump, the long-radius suction elbow inlet illustrated in Figure 1b offers some advantage in reducing the width and depth of excavation under the pump. The inlet velocity to the long-radius elbow, which is usually formed in concrete, is preferably no greater than 3 ft/s (0.9 m/s) at the design flow rate.

A decision to use turning vanes should not be based on a guarantee that there will be an increase in pump efficiency. The design of the vanes—their number and spacing—is still an art more than a science, and it is difficult to prove pump performance in the field. Turning vanes can be effective in eliminating underwater vortices, a problem sometimes associated with suction bells without turning vanes. It has been observed during model testing that the suction bell, as illustrated in Figure 1a, must be placed closer to the back wall than normally recommended for open channel inlets (similar to that in Figure 1j), to prevent underwater vortexing. The flow of water into a closed channel from an open pit containing water of considerably greater depth creates an unequal flow pattern in which the maximum velocity is along the floor. Also, there is little, if any, flow to the back side of the suction bell down from the top of the channel. Unless the bell lip is close to the back wall, flow along the floor and from the front only will overshoot the inlet and roll over, back, and up into the bell, forming an underwater vortex. Although turning vanes can prevent this, they may not prevent uneven flow distribution up into the pump impeller unless they are properly designed. They may even cause hydraulic and mechanical unbalance, which could result in noise, vibration, and accelerated wear of the pump bearings. For this reason, model testing of the turning vanes is recommended.

Screens and Trashracks Although it may not be feasible economically to eliminate all refuse from a pumping system, it probably will be necessary to limit the size and amount of debris or sediment carried into the system. Depending on the probable source of debris, such as a river subject to flooding with considerable flotsam in the runoff and with a very loose bottom, or a lake at constant level without disturbing inlet flows near the structure and with a solid bottom, the protection needed may include only a bar trashrack plus rotating, flushed, fine-mesh screens. If sediment deposit is likely, a settling basin may be required.

The designer must note that, if this equipment is useful, it will pick up debris and gradually increase the velocity through the openings as the net area decreases with blockage. When this occurs at the trashrack, the water level differential will build up, causing a waterfall with increased velocity and turbulence on the pump side of the rack. In addition,

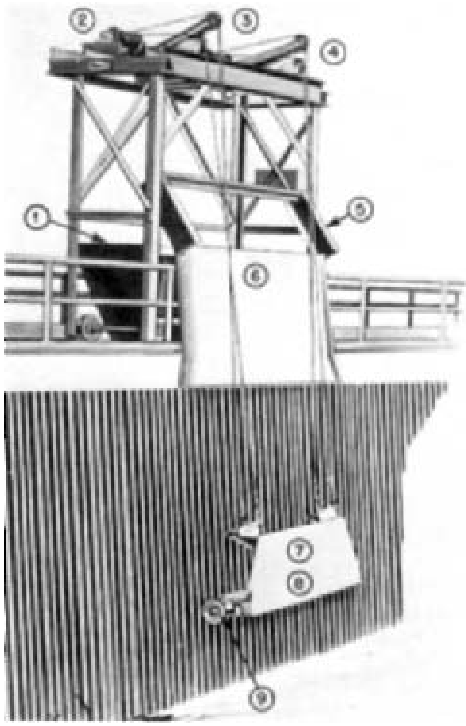


FIGURE 10 Trashrack with raking mechanism: (1) Large enclosed trash hopper contains debris discharged by rake. Hinged door in end of hopper opens wide for debris removal. (2) Heavy-duty, single-drum hoist, push-button-controlled with two separate cables—one for carriage, one for rake teeth. (3) Walking beam actuated by hydraulic cylinder controls position of rake teeth. (4) Fixed sheave. Cable operating over stationary sheave raises and lowers rake carriage. (5) Discharge guide flanges assure positive positioning of rake over trash chute prior to dumping. (6) Dead plate, or apron, integral with superstructure guides rake to discharge point—prevents trash from falling off rake prematurely. Design permits operation over 3.5-ft. (1.1 m) high hand rail. (7) Self-centering rack-guided carriages allows rake to ride over obstructions in water during lowering cycle. Debris of all types picked up on lifting cycle rather than forced to bottom of channel. (8) Rake mechanism assures positive removal of debris with maximum carrying capacity. Hydraulic relief valve provides automatic overload relief. Teeth automatically open if overload occurs, permitting load to drop off rake. No cable failures due to overload. (9) Wide, flanged rollers ride on at least two rack bars (Envirex, Inc., a Rexnord Company)

the increase in velocity may pull more debris through the bars than can be tolerated. It is best to rake these racks (Figure 10) frequently enough to keep the differential head across the rack below 6 in (0.15 m). The spacing of the bars should be such that objects that cannot be pumped would be excluded from passing through. This, in general, will call for the bar spacing to be in proportion to the size of the pump. A pump manufacturer can determine the maximum size sphere a pump will handle, and the bar spacing should be limited to 50% of that value. The size of the bar, the lateral distance between supports, and the pier spacing will influence the rate of debris accumulation and the allowable design differential head.

Rotating screens (Figure 11) will remove trash of a much smaller size because the accumulation is continuously removed and the open area is kept uniform. Finer screening than that required by the pump may be necessary in installations where the liquid pumped must pass through small openings in equipment serviced, such as condenser tubes or spray nozzles (Figure 12). Screens are usually installed in conjunction with trashracks so large, heavy pieces will not have to be handled by the screens. Because velocity through

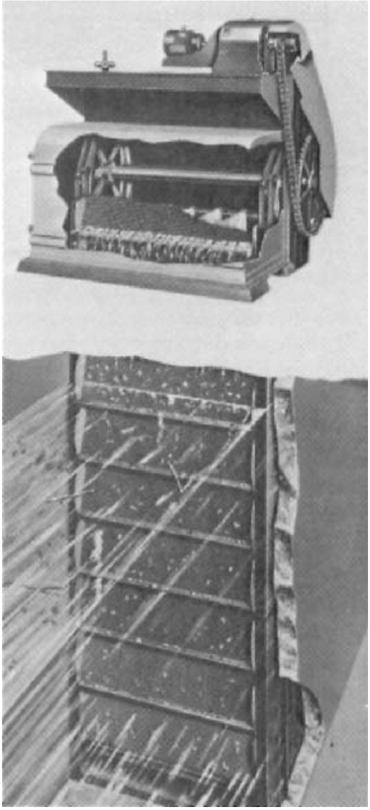


FIGURE 11 Traveling water screen (Envirex Inc., a Rexnord Company)

the screen is limited to 2 ft/s (0.6 m/s) unless environmental considerations require lower velocities, the pit cross section may be determined by screen requirements. If flow is such that a maximum-width screen available would be too long (deep) for practical or economic reasons, two screens may be employed with a center pier. In this case, the distance to the pump should be increased 50% over single-screen distance. Piers should be rounded (radiused) on the upstream side and ogived (tapered to a small radius) on the downstream side. Any corners at the sidewalls should be faired at small angles to the opening and wall to prevent pockets where eddies can form.

The trash collected by the screens must be disposed of. Traveling screens carry trash up into a hood above the operating floor, at which point a series of spray nozzles wash the trash into a trough leading into a disposal area (Figure 13). The nozzles are supplied by pumps sized for 200 to 300 gpm (45 to 68 m^3/h) with pressures of 60 to 100 lb/in^2 (413 to 690 kPa). These pumps are normally deep-well turbine multistage units. They are located in the clear well, if possible, close to a wall. If this is not possible, they can be suspended in the circulating water pit, to one side and ahead of the main pumps. Care must be taken that they do not disturb the flow to the larger pumps. Their submergence requirements are usually less than those of the main pumps, and this allows use of the pump setting that gives the least interference with either pump flow.

It is possible to have these pumps also dewater the pit. This will require additional piping and valves and a more careful location of the pump because it will, of necessity, be close

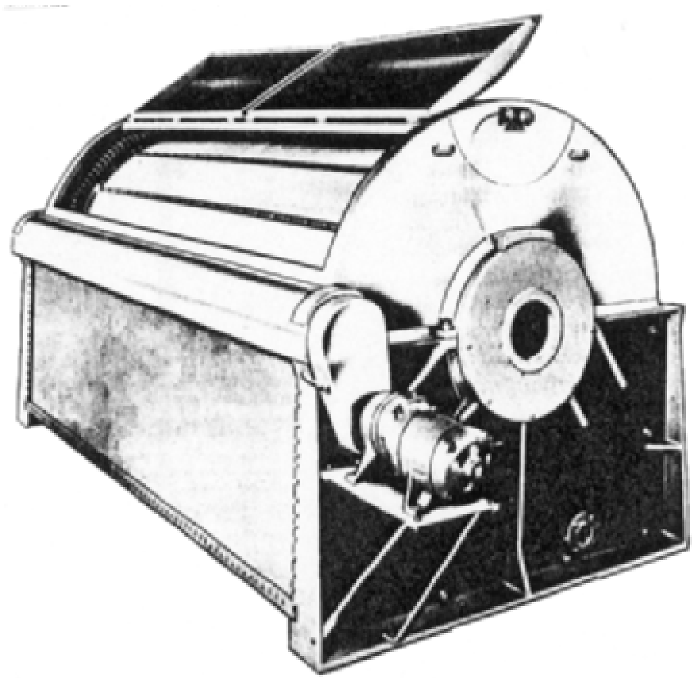


FIGURE 12 Drum filter (Green Bay Foundry and Machine Works)

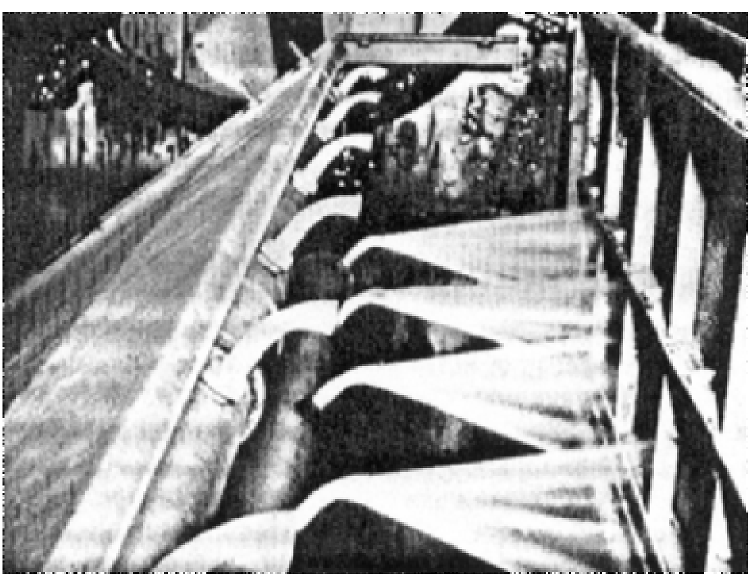


FIGURE 13 Spray nozzle cleaning of baskets on traveling water screens (Envirex Inc., a Rexnord Company)

to the bottom of the pit. A small chamber off the main pit, located far enough from the main pumps to avoid eddying, will be required.

The direction of the flow from the forebay through the screens and into the pump area should be continuous. Avoid right-angle screens, through which flow must change direction at least once and possibly twice. If screens must be at an angle to the flow into the pumps, increase the screen-to-pump distance by 100%. Environmental considerations may increase the possibility of problems in this area.

Environmental Considerations Suction pit requirements will vary according to whether hydraulic or structural standpoints are being considered. Both of these may also be in conflict with environmental considerations.

A design to accommodate fish limitations was mentioned briefly in a previous paragraph. Fish react to a horizontal velocity but are not aware of a pull in a vertical direction. Thus, to keep them from entering the inlet, a horizontal flow must be established at a velocity low enough to permit fish to escape.

Intakes that take their flow directly from a river may have a high velocity that would trap fish. Even if the velocities are lowered to reasonable screen levels—2 ft/s (0.6 m/s)—fish may still be drawn into the screen area and carried up to trash disposal.

When the source of a water supply system is a body of water containing fish, steps must be taken to prevent undue disturbance and destruction of the fish. A site survey should determine

- The intake location furthest from natural feeding areas and from attractive, or "trap," areas
- The number of species involved
- The size range of each species and whether they are anadromous or settled

Sites for intakes should not be selected near feeding areas for large schools of fish (kelp beds, coral reefs, and similar attractive spots). Sheltered spots most suitable for intake flows may also be most attractive to fish.

Next, total flow, probable intake size, and the velocities at inlet, through screens, and at trashracks should be determined. Variations in flow throughout the year and temperature ranges in winter to summer should be available.

The best source of information about local fish is marine biologists who have studied the local areas. They may not only have information on fish habits, feeding patterns, population, and so on, but may also have test information about the fish swimming ability. If they do not already have this information, they can probably run a survey to develop the data.

The most difficult problem to overcome is related to small fish. Screen openings must be held to a minimum, and under velocity conditions, small fish have much less swimming-sustaining ability than larger fish, both in speed and in duration time. In a given steam flow (such as is generated by pumps with inlet water going through screens), a fish must have the ability to sustain a given speed against this flow for a certain length of time. When it weakens, it will fall into the current flow and will be impaled against the screen and destroyed. If the fish senses the velocity early enough, and has an alternate route, it can use darting speed to escape. Or it can follow another attraction (cross velocity flow into a separate chamber or a light attraction to the chamber) and be removed on an elevator or pumped out to a safety channel (Figure 14). Migrating fish need a continuation channel to restore their interrupted journey.

In designing an intake, it is necessary to keep the velocity below 0.5 ft/s (0.15 m/s) through the screen to avoid drawing fish into the screen. For a tube inlet away from shore, a horizontal *velocity cap* (Figure 5) should be placed over the inlet. This will prevent fish from being subject to a vertical velocity and will allow them to maintain a horizontal velocity that will direct them away from the inlet. Alternatively, a cross flow can be created that will propel or attract the fish to one side of the inlet area. From there, they can be directed into a bypass pool and lifted back to their own living area, or they can be sent around the plant to a downstream location. If they are anadromous, they can be sent to an upstream rendezvous. Piers and screens should be kept flush across their inlet face to prevent attractive pockets where fish can hide and be drawn into the screens when they weaken.

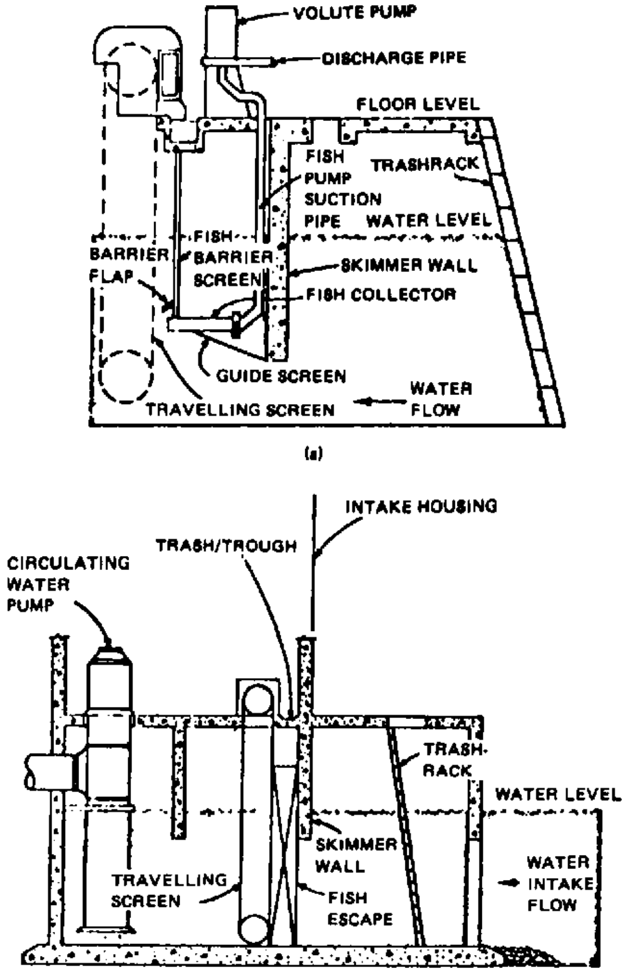


FIGURE 14 (a) Fish pump prevents entrapment on traveling screen (Detroit-Edison). (b) Fish escape allows fish to bypass screen area.

Inlet screen areas should be in small sections rather than one long face so a fish will not be trapped in the center and find it too far to swim to safety after it realizes its predicament. The maximum deterrent flow is about 10 ft/s (3 m/s), but this may be too much disturbance for the flow to the pumps. Smaller areas allow short-term limits for enticing fish away from the inlet, and survival will be much higher.

Fish congregating at an inlet or in a forebay pool can be crowded or herded to an outlet point by the use of vertical nets or horizontal screens. In a direct channel, horizontal moving screens can route fish past a sloping (relative to stream flow, say 35 degrees) moving screen. This results in directing fish to a narrow outlet at one side, leading to an outlet channel away from the main inlet.

The Environmental Protection Agency has had a major impact on plant design since the passage of the Federal Water Control Act of 1972 (as amended) in the US. Enforcement of Section 316a regarding thermal effluent has softened somewhat as later studies indicate that the effects of heat distribution on marine life are variable. Section 316b, however, covers every aspect of best-known technology and is applicable to any facility using a water intake structure.

Young fish must be kept from impinging on inlet screens. This can be accomplished by

- Lowering the velocity through the inlet screens
- Diverting or attracting the fish to other areas
- Providing restraints at inlets
- Using fish buckets and elevators to remove the fish before they can enter the plant area
- In short, by helping the fish avoid contact with the intake to the greatest extent possible

Obviously, trash and fish must be handled in separate areas, and screen wash pressures must be lowered to prevent harm to the fish.

Entrainment of marine organisms too small to be restrained by normal screens causes further problems in areas where such organisms normally develop. If the intake structure must be located in such an area, extensive information must be gathered at the site. Analyzing as much data as are available, in conjunction with plant flow and location requirements, may give the designer some idea of which equipment to select.

Small slot-width wedgewire screens (Figure 15) are now being furnished in larger sizes, and, with their very low inlet velocity and backflushing capability, offer one good approach to reducing entrainment. Sand, chemical, and cloth filters may suit some situations, but backflushing and cleaning problems make them less attractive costwise.

Equipment may consist of any combinations of

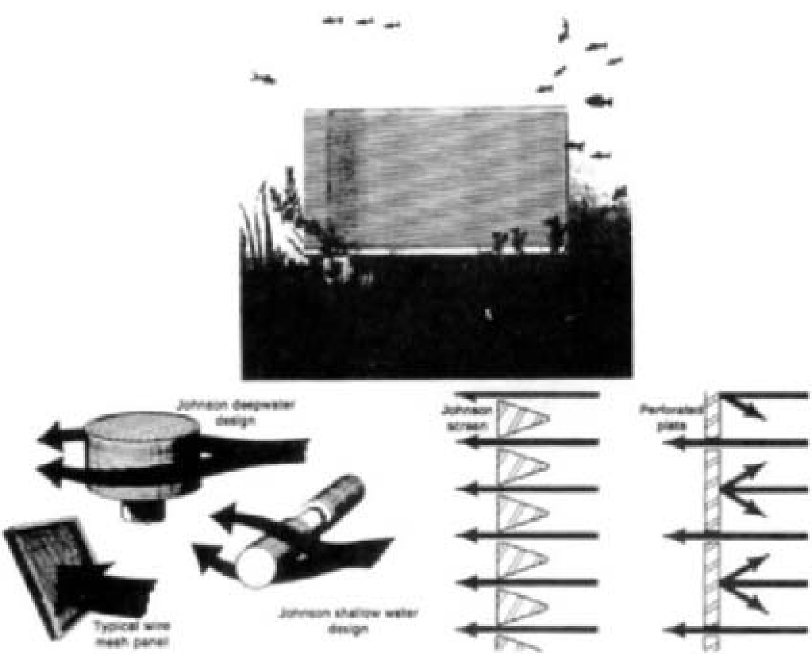


FIGURE 15 V-shaped slotted screen provides velocity control to avoid attracting fish and good air or water backflushing for debris cleanup (Johnson Division, UOP).

1. Stationary screens
   a. Bar racks in various attitudes (Figure 10), with or without automatic rakes
   b. Air bubble, lighting, or electric heaters
   c. Underground, as in Ranney wells (Figure 6)
2. Moving screens
   a. Vertical or horizontal (Figure 11)
   b. Drum (Figure 12)
3. Transportation devices
   a. Elevators
   b. Pumps (Figure 14)
   c. Baskets on vertical screens
   d. Attractive escape areas (Figure 15)
   e. Velocity changes
4. Remote intakes
   a. Ocean outfall with velocity cap (Figure 5)
   b. Ranney well (Figure 6)

If the source is a river, the angle of the intake structure relative to the direction of flow is important in modifying the impact of these design requirements. A case in point is shown in Figure 16. The screen house at a low angle to the river flow (instead of the usual 90 degree inlet) allows the river current to provide a swim-by attitude for fish while a low-velocity screen approach is maintained. If the pump house is in line with the screen house,

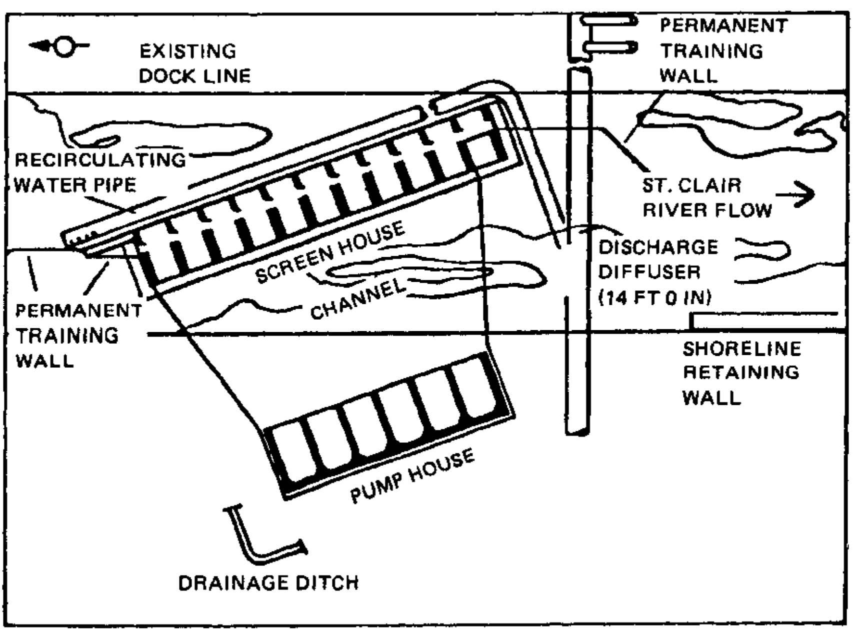


FIGURE 16 Angling the intake structure to the river flow allows fish to swim by Belle River power plant (Detroit-Edison)

minimal disturbance will be felt by the pumps. A tendency for eddies to form (and create vortices) can be minimized by placing wing walls upstream and downstream to control velocity.

Testing Model Pit Design When the basic rules for good pump suction pit design are adhered to, no model test will be required to ensure proper operation of the pumps and pumping system. The substance of these rules is to keep a straight-in approach at a constant low velocity from the water source to the pump chamber. The ANSI/Hydraulic Institute Standard dimensions and charts satisfy these criteria for the average pump in general application.

Site layout problems may make the ideal solution impossible. Structural and environmental requirements may outweigh hydraulic requirements in some instances. When the ideal pit may not be possible or economically feasible, a model test should be considered. It should be noted that the ideal dimensions are a composite covering not only a range of specific speeds but also a complex melding of pump design philosophies. Some variation from ideal dimensions should be expected from individual pump manufacturers.

Pump manufacturers are not in a position to guarantee the pump pit design. Differences of opinion between the structural and hydraulic pit design engineers and the pump design engineers may best be resolved by performing a model pit test (Figure 17). Refer also to Section 10.2.

Vortexing The real problems resulting from improper pit design occur largely on the water surface in the form of vortices, or cones, produced by localized eddies on the surface of the water. If this disturbance continues, the flow of water will carry the underwater part of the vortex down toward the pump suction bell and ultimately into the pump (Figure 18). This introduces air into the impeller and will affect the mechanical radial balance of the impeller by interrupting the normal solid-liquid flow pattern. This type of disturbance will produce hydraulic pulsations in the pump flow and mechanical overloading of bearings and impeller guides.

Underwater vortexing sometimes occurs in round pits or in pits where the pump suction bell is at some distance from the rear wall. Flow past the suction bell strikes the rear

FIGURE 17 Model pit test setup with fixed screens and pumps and valves for variable flow, to scale (Flowserve Corporation)

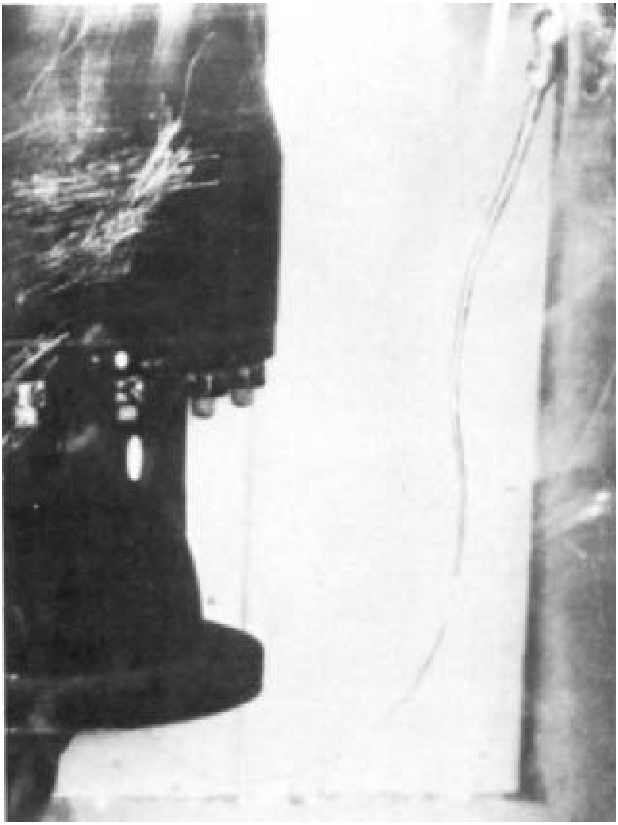


FIGURE 18 Surface vortex drawing air into pump suction bell

wall and rolls back toward the bell, forming an eddy current that disturbs the normal flow into the pump. In a round pit, a cross baffle below the pump bell may reduce this effect. Where the pump is some distance from the back wall, a wall can be installed near the pump, or a horizontal baffle at suction bell level behind the pump will also reduce the disturbance. In all cases, the distance between the suction bell and the bottom of the pit should not be more than one-half the suction bell diameter, and one-third the diameter is preferable.

The use of the suction bell diameter as a basis for spacing should be carefully evaluated. It will be seen that there is nothing magical in this relationship, especially when several pump manufacturers all use different bell diameters. The real criterion is the allowable velocity at the suction bell. It has been found that very-low-head pumps are much more sensitive to bell velocity over 6 ft/s (1.8 m/s). For example, the velocity head loss at the bell inlet with a high velocity may be such a large percentage of total pump head that efficiency could drop as much as 10%. A good design rule for safe operation can be related to pump head. For pumps having up to a 15-ft (4.6-m) head, the suction bell velocity should be held to 2.5 ft/s (0.76 m/s); up to 50-ft (15-m) head, 4 ft/s (1.2 m/s); and above 50-ft (15-m) head, 5.5 ft/s (1.7 m/s). These values should be used for any substantial amount of pumping, but for occasional short-term pumping, they can be exceeded without destroying the pump.

Vortices may be broken up and effectually nullified by arrangements of baffles and vanes, or they may be prevented from occurring initially by a proper pit design. The only way to determine what type of baffling should be used and its effectiveness is by model testing. Methods of eliminating vortexing are discussed in Section 10.2.

Vortices are usually generated when the flow direction of the liquid to be pumped changes or when there is high velocity past an obstruction, such as a gate inlet corner, screen pier, or dividing wall. In combination, these two causes invariably generate vortexes. For this reason, the pump suction pit should be immediately preceded by a straight channel in which the velocity does not exceed 1.25 ft/s (0.4 m/s). Satisfactorily operating pump pits with higher velocities are rare and should not be put into operation without the assurance of a model study.

An additional condition likely to generate vortexes is a multiple pump pit with individual cells in which only a portion of the pumps will operate simultaneously. The dead space behind the non-operating pump will have flowing water tending to reverse direction and form eddies. Eddies and vortices can be avoided by eliminating or venting the walls at the rear of the pit. It also helps to position the pumps at the extreme rear of the pit. Alternatively, expensive modifications to the pit, such as splitter walls and baffles, may be required (Figure 7).

Round pits tend to generate vortices, especially when the pump is centered in the pit. These vortices usually occur around the pump column because of eccentric inlet flow. Special cases of the round pit are tolerable either when a Ranney well (Figure 6) is used or when there are booster pumps in the pipelines (Figure 19). In the Ranney well, the ratio of pump size (and flow) to pit size (and capacity) is such that very low velocity exists, as in a lake inlet. Water comes into the Ranney well all around the periphery. These conditions of direct flow and low velocity prevent vortexing. Booster pumps installed in a circular can

FIGURE 19 Booster pump suspended in a steel well (or can) that requires a minimum space for suction pit (Flowserve Corporation)

(suction tank) must be centered in the can, and all inlet velocities to the can and flow in the can and into the pump must be uniform and high enough to provide fluid control. This velocity will vary from 4 to 6.5 ft/s (0.76 to 1.98 m/s).

Vortices are not generated by a pump or pump impeller and so do not fall into clockwise or counterclockwise rotation because of the pump rotation. Also, in a pump pit, vortices do not have a directional rotation induced by the rotation of the earth and therefore are not of opposite rotations above and below the equator, as is the case with "bathtub vortexing," which occurs in tanks being drained without pumps.

Submergence Centrifugal pumps in intake pumps must be submerged deeply enough to provide

- A pressure sufficient to prevent cavitation in pump first-stage impellers, referred to as *NPSH* (it is assumed that the proper pump has been selected to perform satisfactorily with available *NPSH*)
- Prevention of vortexing and associated pit flow problems detrimental to pump operation

A pump may have adequate submergence from a pressure standpoint and still be lacking in sufficient depth of cover above the suction inlet to prevent surface air from being drawn in. Any wet-pit pump must have its suction inlet submerged at all times, and for continuous pumping every pump will have a fixed minimum submergence requirement. Because this relates to velocity, there are two basic parameters: the suction inlet diameter and the depth of water above the inlet lip. As pump size (and flow) increase, the inlet velocity may stay constant as the bell diameter increases, but at the same time, the impeller distance above the suction inlet becomes larger, so a fixed submergence value would lead to increased surface velocity, peripheral drawdown, and an increase in air intake.

Basically, submergence must of necessity increase with pump size. For the final determination, some balance must be struck between submergence and pit width to satisfy an average flow velocity of 1 to 1.25 ft/s (0.3 to 0.4 m/s) and maintain reasonable economic balance between excavation costs, concrete costs, and screen costs without neglecting ecological requirements and still fulfilling the primary need for circulating water in adequate quantities.

SUCTION TANKS

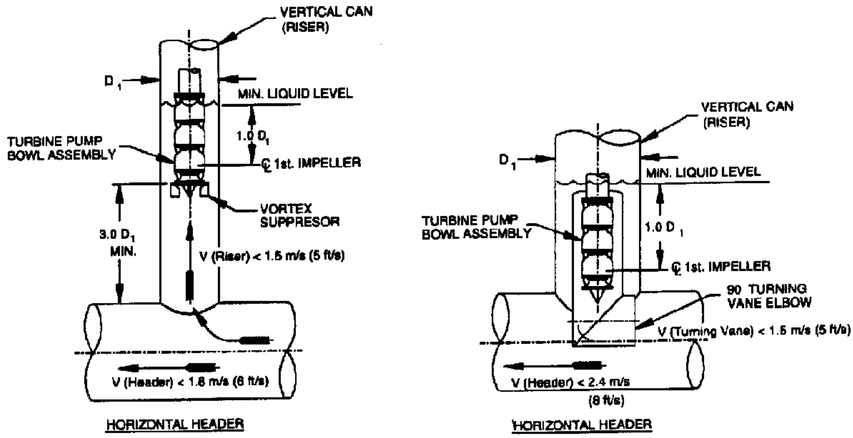
There are a variety of shapes and configurations for suction tanks. They may be vertical or horizontal, cylindrical, or rectangular. Outlets from the tanks may be vertically downward, horizontal, from either the side or bottom of the tank, or in the case of a pressurized tank, vertically upward. The formation of vortices in the tank can cause air or gas entrapment even when the tank outlet is fully submerged. This can cause pulsating pump flow, noisy operation or a reduction in pump performance.

To determine minimum submergence of the outlet pipe in the tank, the Hydraulic Institute recommends the following relationship:

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

where F_D = Froude number $= V/(gD)^{0.5}$
D = outlet fitting diameter, ft (m)
S = submergence, ft (m)
V = outlet fitting velocity, ft/s (m/s)
g = acceleration of gravity, ft/s^2 (m/s^2)

Figure 20 shows examples of how to apply the calculated submergence value depending on tank orientation and outlet configuration. Figure 21 shows how to use the above equation to calculate velocity for the Froude number calculation, depending on type of



a) no turning vanes b) turning vane elbow

FIGURE 20 Open bottom can intakes for pumps less than 5000 gpm (315 l/s) (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

outlet fitting and direction of tank outlet flow and nozzles. If the tank has multiple inlets or outlets, the design should be such that flow interaction is not detrimental to overall performance.

CAN PUMP INTAKES

A can pump is defined as one that has a can, or barrel, surrounding the pumping unit. This can acts as a "sump" or intake structure for the pump suction impeller. The can may be either closed bottom, and contain the pump suction nozzle, or open bottom and connect directly to a piping header. The can design must provide uniform, stable flow distribution to the suction impeller inlet.

Vertical turbine pumps (Figure 19) require uniform inflow to the suction bell to avoid swirling and submerged vortices that may result in cavitation, vibration, and accelerated pump wear. When a pump with an open bottom can design is connected to a horizontal header (Figure 20a), the velocity in the header should be no more than 6 ft/s (1.8 m/s) to allow the liquid to turn and flow upward to the pump suction bell. The velocity in the can rising to the pump should not exceed 5 ft/s (1.5 m/s), and the suction bell inlet should be located at least three diameters above the top of the horizontal header. For velocities approaching the maximum recommended levels, vortex-suppressing vanes may be added to the suction bell area to break up swirling and nonsymmetrical flow patterns as they approach the impeller inlet.

If a 90 degree turning vane elbow on the can assembly surrounding the pumping element is used (Figure 20b), the velocity in the horizontal header can be as high as 8 ft/s (2.4 m/s). The turning vane elbow should be sized for a maximum velocity of 5 ft/s (1.5 m/s).

Most can pumps are of the closed bottom design (Figure 21). In this arrangement, the pump suction nozzle is located either in the can or in the pump nozzle head that contains both the suction and discharge nozzles. The pumping element (assembly) must be centered in the can to avoid non-uniform flow to the suction impeller. Flow straightening vanes are suggested for all can intakes. Because of the limited volume of liquid in the can, surging of the liquid level within the can—or barrel—cannot be tolerated; the can should be

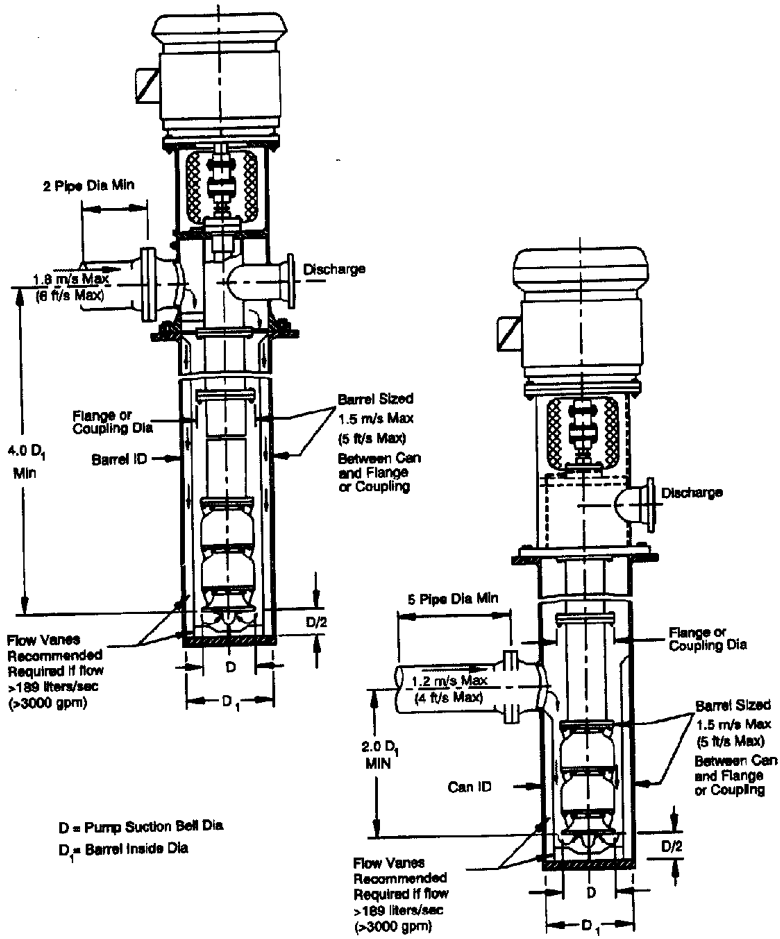


FIGURE 21 Closed bottom can (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

arranged so it is always full. The velocity of the flow between the pumping element and the inside of the can should not exceed 5 ft/s (1.5 m/s). For a suction nozzle velocity of 4 ft/s (1.2 m/s) maximum, the centerline of the nozzle should be at least two diameters above the suction bell inlet. To ensure uniform flow distribution into the can, the suction nozzle should be connected to at least five pipe diameters of straight pipe before elbows or other flow-disturbing fittings are installed.

Pumps with submersible well-type motors require flow around the motor for cooling. A shroud is typically used to direct flow across the motor as it goes into the pump. The top of such a cooling shroud is covered to restrict downward fluid flow and still allow venting of air from the shroud. This arrangement is described in detail in ANSI/HI 9.8-1998 (Reference 1). It is recommended that the inlet piping be sized to limit draw-down of the liquid below the minimum required level during the startup to a period of less than 3 seconds.

SUCTION PIPING

Single Pumps Piping to the suction of a dry-pit centrifugal pump (Figure 22) must be carefully planned to provide uniform, straight-line flow to the impeller, and adequate pressure and sealing against leakage, in or out. Air pockets just prior to entrance to the pump should be avoided, as well as down flow lines subject to sudden pressure changes. Air pock-

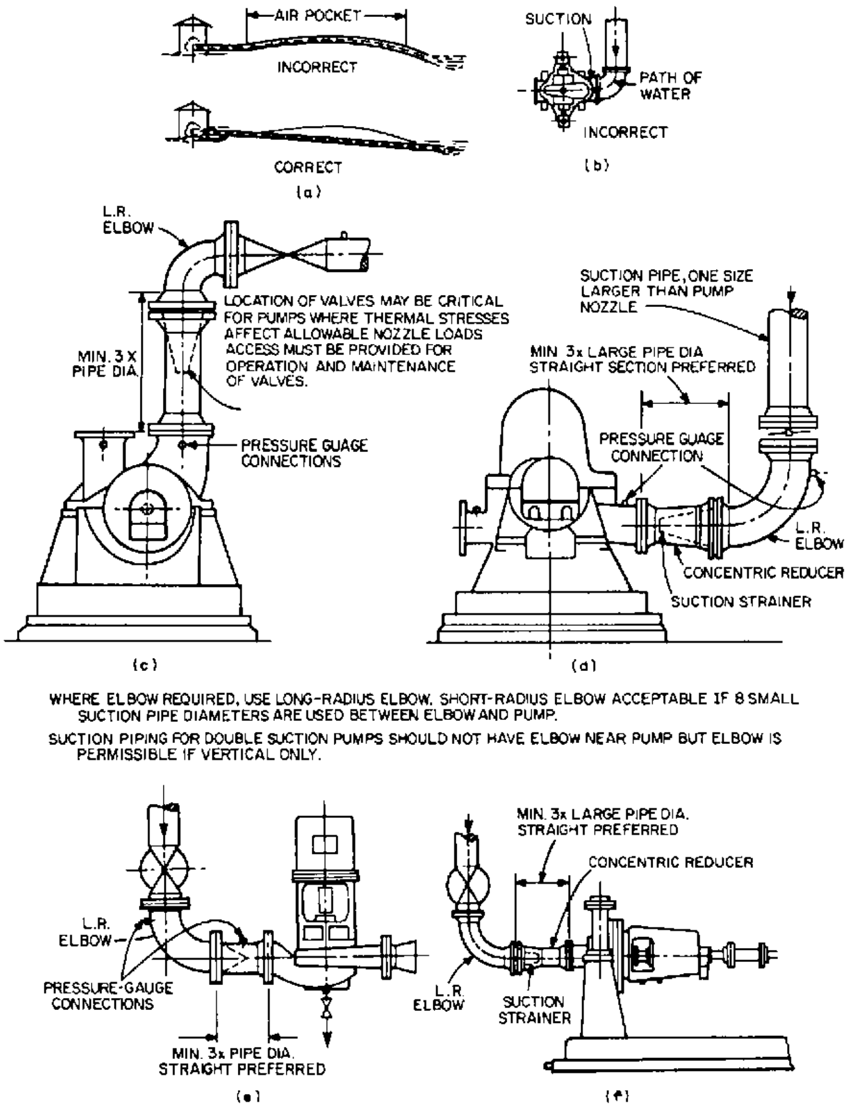


FIGURE 22 Suction piping faults for dry-pit centrifugal pump. (a) Air pockets should be avoided. (b) Suction elbow should not be in a plane parallel to pump shaft. (c) Valve location may be critical to pump nozzle loading. (d), (e), Suction elbow should be one size larger than pump's, long radius and three large pipe diameters distant. (d), (f) Concentric reducer should be installed distant from pump.

ets can be prevented by proper elevations (Figure 22a). Pressure surges can be controlled by surge tanks, air tubes, and so on, which may require a system pulsation study to determine possible need and solution.

To provide an optimum flow pattern to avoid impeller disturbance, it may be necessary to have a straight run of pipe of as much as eight pipe diameters immediately prior to the pump suction (for example, following a short radius elbow or tee). Following a long radius elbow or a concentric reducer, a straight run of at least three pipe diameters is recommended (Figure 22c to f). Eccentric reducers should not be used next to the pump suction nozzle. Although installing eccentric reducers with the flat side on top will eliminate a potential air pocket, large changes in diameter could result in a disturbed flow pattern to the impeller and cause vibration and rapid wear. Pipe venting, in conjunction with a concentric reducer, may be preferable to the use of an eccentric reducer.

Ideally, a suction pipe should approach a double suction pump perpendicular to the shaft centerline. If there is an elbow in the suction piping upstream of the suction flange, it should be bringing flow from either overhead or below, not from the side of the pump. If there is a short radius elbow or other flow-disturbing device in the suction piping upstream of the suction flange, there should be at least five pipe diameters between the device and the suction flange. If a short radius elbow is in the same plane as the impeller shaft, there should be at least eight pipe diameters between the elbow and the suction flange. An incorrect installation could result in an uneven flow to both sides of the double suction impeller. This could cause a reduction in capacity and efficiency, an increase in thrust on the bearing, noise, and possible cavitation damage to the impeller.

A dry-pit pump (Figure 1g) may operate with a suction lift and therefore will be located above the liquid source. All losses in piping and fittings will reduce the available suction pressure. Suction piping should be kept as simple and straightforward as possible. Any pipe flange joint or threaded connection on the suction line should be gasketed or sealed to prevent air in-leakage, which would upset the vacuum and keep the pump from operating properly.

If expansion joints are required at the suction of a pump, an anchor should be interposed between the pump nozzle and the expansion joint to prevent additional forces from being transmitted to the pump case and disturbing rotating clearances. The same requirement applies to a sleeve coupling used to facilitate installation alignment.

Reciprocating pumps must have additional consideration because of the pulsating nature of their flow. Suction piping should be as short as possible and have as few turns as possible. Elbows should be long-radius. Pipe should be large enough to keep the velocity between 1 and 2 ft/s (0.3 and 0.6 m/s). This will generally result in pipe one to two sizes larger than the pump nozzle. High points that may collect vapor are to be avoided or, if necessary, properly vented. A pulsation dampener or suction bottle should be installed next to the pump inlet. Available *NPSH* should be sufficient to cover not only reciprocating pump requirements and frictional losses but also acceleration head (see “Surge and Vibration”).

Manifold Systems All comments relative to single pumps apply to manifold-pump systems, as well as some additional points.

In a suction manifold, the main-line flow should not be more than 3 ft/s (0.9 m/s). Branch outlets should be at 30 to 45 degrees relative to main-line flow rather than 90 degrees, and velocity can increase to 5 ft/s (1.5 m/s) through a reducer (Figure 23). With such velocities, branch outlets can be spaced to suit pump dimensions in order to avoid crowding. Also, if the angled manifold outlet is used, pumps can be set as close to the manifold as the elbow, valve, and tapered reducer will allow.

Manifold sections beyond each branch takeoff should be reduced to such a size that the velocity remains constant. One exception to this scheme is a tunnel suction (Figure 24). The flow through the tunnel may operate independent of the pumps, which are suspended in boreholes drilled into the roof of the tunnel. Boreholes at least one-third of the tunnel diameter should be horizontally spaced at least 12 borehole diameters apart. Smaller ratios can be closer to a minimum of six diameters. Pump suction bells should be at least two borehole diameters above the tunnel roof. Velocities in the tunnel should be kept below 8 ft/s (2.4 m/s) for best pump performance.

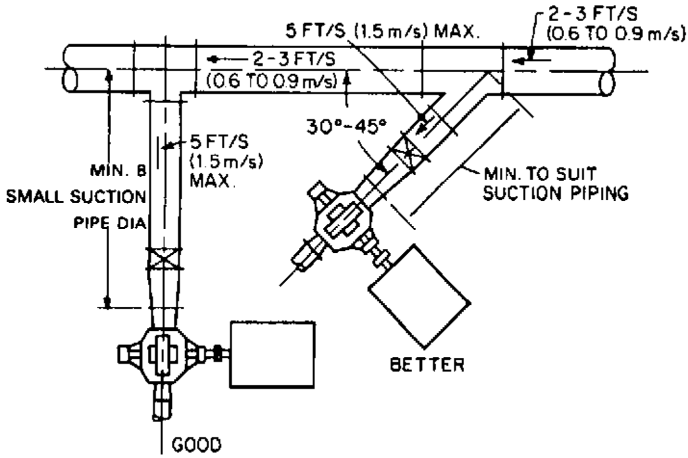


FIGURE 23 Suction pipe header recommendations for dry-pit centrifugal pump

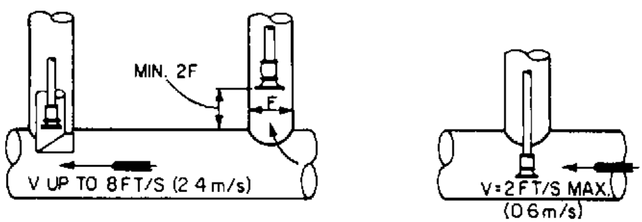


FIGURE 24 Higher tunnel velocities require isolation of pump from direct flow to prevent distortion of close-clearance parts and shaft (Hydraulic Institute Standards, 13th Edition, 1975—out of print)

NPSH The net positive suction head so essential to trouble-free pump operation is always reduced by losses in suction piping. An economic balance must be obtained between pump size and speed, required *NPSH,* pipe size, and suction velocity. If the suction source is a tank, such as a deaerator in a power plant, it is quite expensive to elevate the tank. Therefore, the available *NPSH* is low. The pipe size from the tank to the pump should be large for low velocity and the length should be short for minimum losses. A cooling tower on a hill supplying water to a pump station below could have a much higher pipe velocity and longer length without forcing an extremely low *NPSH* requirement on the pump.

A dry-pit pump should be as close to the suction source as possible. When the *NPSH* required indicates a suction lift is possible, the most advantageous solution is to reduce the suction losses by increasing the pipe size, flaring the inlet bell, and keeping the pump suction eye of the first-stage impeller close to the minimum water level.

High-Pressure Inlets Pumps in series build up pressure, so the second and following pumps will have a high-pressure suction piping connection. This emphasizes the need for tight joints and flanges and careful welding. Expansion joints should not be used because the hydraulic forces on the pump would be large, difficult to restrain, and perhaps impossible for the pump to handle without distortion. As *NPSH* will not be a problem for pumps downstream of the first pump, the pipe size between pumps can be kept small to minimize design problems and valve costs.

Effect on Pump Efficiency Other than mechanical operation, the greatest effect of flow disturbance at the pump suction is on pump efficiency. The higher the pumping head, the lesser this effect becomes. For very high-head pumps, a high velocity may be ignored completely unless there is an extremely large power evaluation factor. Greater attention should be given to the suction pipe design for pumps producing 100 ft (30 m) of head or less than those with high head, as efficiency may be worth many dollars in power costs over the life of the plant and equipment.

Pump efficiency relates to the efficiency of the whole system, and so it may be well worthwhile to invest more money in a large suction pipe.

Surge and Vibration One of the possibilities arising from a power failure at a pump station is the reversal of a centrifugal pump if a valve fails to close, and its subsequent operation as a turbine. Under rated head, a pump will run from 20 to 60% above rated speed in the reverse direction. In its transition to that phase, the forward motion of the water is interrupted and gradually reversed. At the time of the power failure, the flow velocity in the suction pipe may not decelerate slowly enough to prevent a surge in the direction of the pump. The suction piping should be designed to withstand the resulting pressure rise because absolute integrity of valving and power supply will be too costly as a design parameter (see Chapter 8 for additional information on this subject).

Rotating machinery must have a design vibration frequency that is sufficiently removed from a system frequency to avoid sympathetic activity. Any combination of vibration frequencies near enough to each other to react will do so when a prime source, such as a pump, excites them. When a piping system has been designed to allow only very low stresses to be transmitted to a pump nozzle, it is quite vulnerable to vibration. It is usually good practice to analyze a suction system made up of pipe, valves, hangers, restraints, pump nozzle loads, pump speed and impeller configuration, foundation, and anchors to be sure the system is not "in tune." At the design stage, it is relatively easy to change a valve or elbow location or to add a surge suppresser. It is usually much more costly to change a pump.

Reciprocating pumps are surge producing machines. In particular, they require sufficient energy at suction to overcome pump required *NPSH* and pipe friction and a form of energy called *acceleration head*. The pump energy must overcome the acceleration-deceleration pulsation flow in the suction end, which could lead to liquid flashing with pump noise and vibration. Surges large enough to rupture the pump cylinders may also be produced.

The total *NPSH* required by a reciprocating pump must include the *NPSH* required by the pump plus frictional loss from the suction pipe plus acceleration head. Acceleration head in feet (meters) is

$$Ha = \frac{LVNC}{gK}$$

where L = actual (not equivalent) length of suction pipe, ft (m)
V = velocity of flow in the suction line, ft/s (m/s)
N = pump speed, rpm
C = pump constant, decreasing with number of cylinders from 0.2 to 0.04
g = acceleration of gravity, 32.17 ft/s^2 (9.807 m/s^2)
K = liquid constant: 2.5 for compressible fluids, 1.5 for water

The acceleration head can be reduced by installing a pulsation dampener in the suction line near the pump.

PIPE SCREENS AND STRAINERS

Except for special designs for handling slurries and other solids, most pumps are limited in their ability to handle dirt, trash, and solids of any size down to microparticles. Pump passageways (volutes, diffusers, impellers) handle generally spherical objects up to a "sticking size," but a severe problem may be caused by dust-size particles that lodge in sleeve bearings, bushings, and wear rings and cause either rapid wear or rotation seizure. A wet-pit source may never be quite free from some degree of silt or sand. After it is accepted by the pump, this liquid goes into a piping system, usually one with other pumps involved. The original pit pump may also be a dry-pit type. In either case, if the pumps of a system are to operate for long periods of time with minimum maintenance, wearing parts must be protected by fine screens or strainers.

In any screening operation, it is essential to remember that if the screen is necessary and does a good job, it will plug up and thus requires cleaning. In a wet pit, therefore, neither the suction bell of a wet-pit pump nor the suction bell going to a dry-pit pump should be basket-screened. Collection of debris will soon increase inlet velocity to unmanageable proportions, and the pump operation will suffer.

It follows that if the basic screening discussed previously does not provide liquid clear enough to be pumped, additional strainers must be put into the system. The need for such strainers, and the degree of particle reduction, are usually specified by the pump manufacturer. In some instances, the system may require water as pure as it is possible to produce, as in boiler-feed service. For any situation, a variety of strainers is available.

Drum rotating strainers (Figure 12) can handle solids as small as approximately 0.01 in (250 μm). Backflushing is used to renew the strainer area. The addition of woven wire cloth (Table 2) to such strainers may remove materials down to 0.1 in (25 μm). Woven wore drum strainers used in conjunction with sand filters will reduce the amount of backflushing required for the sand filters.

Where a finer degree of filtration is required, a line filter is used. Single flow filters require system shutdown to remove and clean baskets, but either a motorized automatic type of single flow filter (Figure 25) or twin filters with alternate flow by valve arrangement (Figure 26) permit uninterrupted system operation. When a particularly difficult situation exists, a battery of filters in series may be used.

For areas where wear can be extremely critical, such as bearings and stuffing boxes (mechanical seals), additional filtering is necessary, especially if the entrained silt is abrasive. A vortex filter using centrifugal flow action will remove particles down to a few micrometers. These filters have continuous sediment removal and should not require any maintenance (Subsection 2.2.1, Figure 78).

Some systems, such as boiler feedwater, are "closed" after they are in an operating cycle. Chemical and mechanical cleaning components are designed into the system to obtain a high degree of water purity. Before the system is ready to operate, however, the piping, tanks, valves, and so on, will have a residue of weld spatter, metal chips, and other debris from the fabrication process. Temporary strainers are used in the suction lines of boiler feed pumps and condensate pumps to gather up this debris. These strainers are in-line fabricated metal mesh of either a cone or box type. Feed pumps are most often equipped with cone strainers, pointing preferably upstream (Figure 27). The length and diameter are set to conform to the suction pipe size and the mesh so the velocity does not become excessive. These strainers, set between flanges in a pipe section, must be removed and cleaned whenever the differential pressure reaches 2–3 lb/in^2 (15 to 20 kPa). Condensate pump strainers are usually box strainers, a square section directly in front of the suction flange with a square box larger than the pipe diameter laced with egg crate metal mesh (Figure 28). This basket strainer can be lifted out of the box for cleaning as necessary. These strainers for collecting fabrication debris may be removed when the need disappears and the system will remain clean.

TABLE 2 Wire mesh data

Screen mesh number, square weave	Equivalent in corduroy weave	Size		% open area	Screen mesh number, square weave	Equivalent in corduroy weave	Size		% open area
		in	μm				in	μm	
6		0.126	3327	57.2	90	20 × 120	0.0056	159	25.4
8		0.090	2362	51.8	100	20 × 150	0.0055	147	30.3
10		0.068	1651	46.2	115		0.0051	124	30.0
12		0.060	1397	51.8	120		0.0046	120	30.7
14		0.051	1168	51.0	130		0.0043	115	31.1
16		0.045	991	53.0	140		0.0042	109	34.9
20		0.033	833	43.6	150		0.0041	104	37.4
24		0.0287	701	47.4	160	20 × 200	0.0038	96	36.4
28		0.0227	589	40.4	170	20 × 250	0.0035	88	35.1
30		0.0203	495	37.1	180	20 × 300	0.0033	82	34.7
35		0.0176	417	37.9	200	20 × 350	0.0029	74	33.6
40	12 × 64	0.0150	380	36.0					
42		0.0138	351	33.6					
50		0.0105	280	27.6					
60	14 × 88	0.0077	246	21.3	120 × 330			70	
65		0.0084	208	29.8	120 × 400			40	
70		0.0068	190	22.7	120 × 600			30	
80	24 × 110	0.0070	175	31.4	200 × 600			25	

1 μm = 10^{-6} m = 1 micron = 0.00004 in; 1 in = 25.4 mm.
SOURCE: Green Bay Foundry & Machine Works.

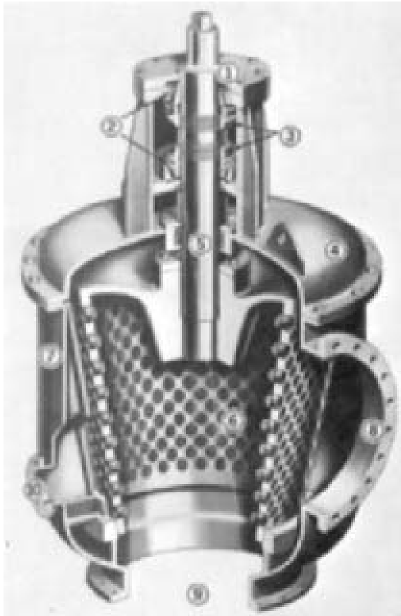


FIGURE 25 Single flow strainer with provision for backwashing without interrupting service: (1) strainer drive, (2) strainer drum support bearings, (3) lock nuts for drum adjustment, (4) cover, (5) shaft, (6) drum—tapered for adjustment, (7) body, (8) inlet, (9) outlet, (10) backwash opening (S. P. Kinney Engineers)

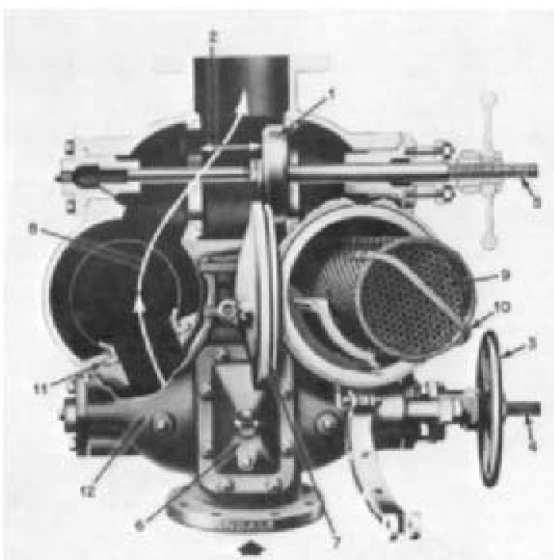
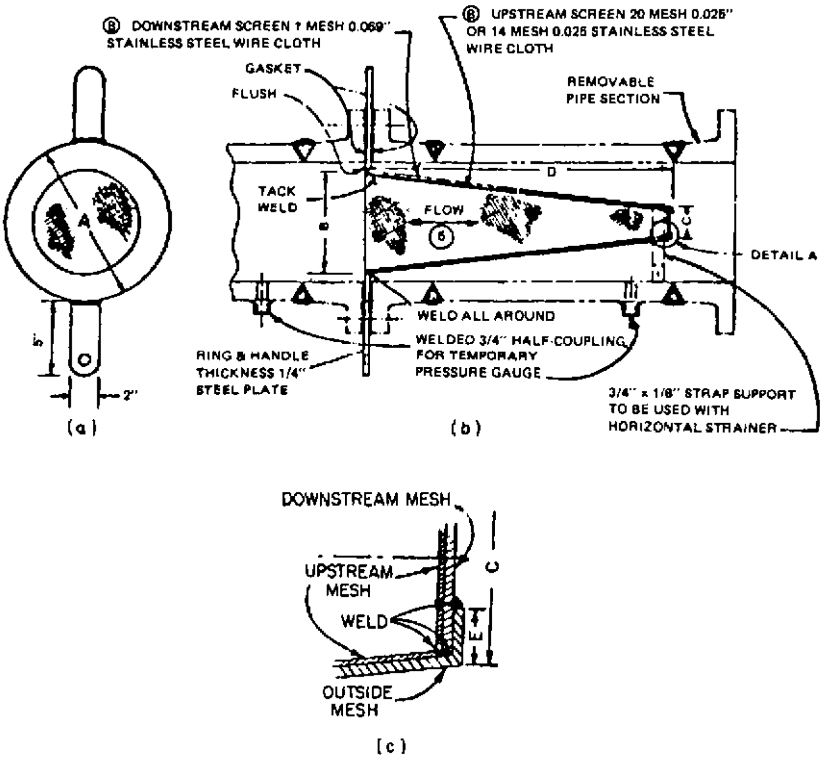


FIGURE 26 Duplex basket strainer. Flow through one side allows cleaning of opposite basket while line is still in service. (1) positive flow diversion, (2) rapid switchover, (3) free-turning handwheels, (4) visual indication of valve position, (5) external acme threads, (6) internal access, (7) cover and clamp(s), (8) streamlined flow, (9) basket, (10) basket handle, (11) basket, (12) one-piece body construction (Andale)



Flange or pipe size	A	B	C	D: 20 mesh 0.025-in wire, 0.025-in opening	D: 14 mesh 0.025-in wire, 0.046-in opening	E
3	5	2-3/4	1-1/2	16	11	1/4
4	6-3/16	3-1/4	1-1/2	16	11	1/4
6	8-1/2	5	2	18	12	1/2
8	10-5/8	7-1/8	2	20	13	1/2
10	12-3/4	9-1/8	3	23	14	1/2
12	15	11	3-1/8	24	15	1/2

FIGURE 27 Cone strainer for start-up service in boiler-feed pump system. (a) cross-section, (b) lengthwise section, (c) typical detail A (nose pointing downstream). All dimensions are in inches (1 in = 0.0254 m = 2.54 cm). Strainer ring material is carbon steel. When a strainer has served its purpose, the screen portion is removed, the ID of ring is cut to suit ID of pipe, and the ring is used as a spacer. Cone strainers may be installed with nose pointing either upstream or downstream. The fine mesh screen should always be on the upstream surface of the strainer.

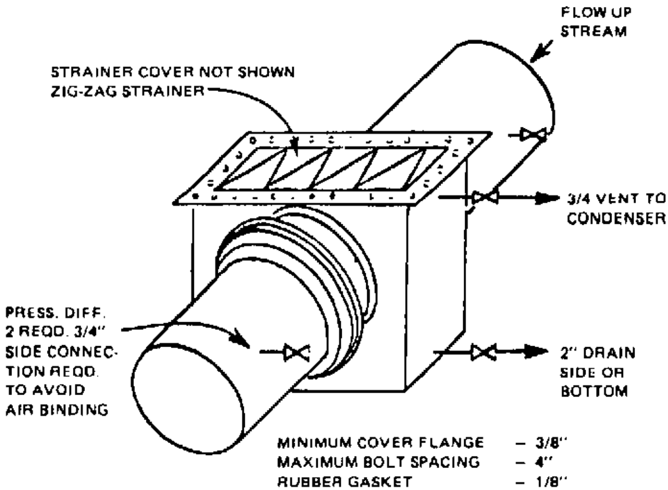


FIGURE 28 Box strainer for temporary start-up service. Pressure differential connections are on top or side only. Upstream mesh is #16 stainless steel with 0.018-in (0.46-mm) wire diameter. Downstream mesh is #2 stainless steel with 0.063-in (1.6-mm) wire diameter (1 in = 25.4 mm).

REFERENCES

1. American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Hydraulic Institute, Parsippany, NJ, www.pumps.org.

FURTHER READING

Aten, R. E. "Alternative Cooling Water Sources for the 'Water-Short' Midwestern States." Paper presented at American Power Conference, Chicago, IL, 1979.

Budris, A. R., and Mayleben, P. A. "Effects of Entrained Air, NPSH Margin, and Suction Piping on Cavitation in Centrifugal Pumps." 15th International Pump Users Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, TX, 1998.

Dicmas, J. L. "Development of an Optimum Sump Design for Propeller and Mixed Flow Pumps." Paper presented at ASME meeting, New York, 1967.

Dicmas, J. L. "Effect of Intake Structure Modifications on the Hydraulic Performance of a Mixed Flow Pump." *Joint Symposium on Design and Operation of Fluid Machinery.* Vol. 1, p. 403, June, 1978.

Dornaus, W. L. "Flow Characteristics of a Multiple-Cell Pump Basin." *Trans. ASME*, July 1958, p. 1129.

Dornaus, W. L. "Stop Pump Problems Before They Begin with Proper Pit Design." *Power Engineering Magazine,* February 1960, p. 89.

Fraser, W. H. "Hydraulic Problems Encountered in Intake Structures of Vertical Wet Pit Pumps and Methods Leading to Their Solution." *Trans. ASME*, May 1973, p. 3.

Howard, J. H. G., and Hermann, P. "Flow Field Investigation for a Series of Right-Angle Minimum Axial Depth Pump Intakes." 2nd International Pump Users Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, TX, 1985.

Langewis, C., Jr., and Gleeson, C. W. “Practical Hydraulics of Positive Displacement Pumps for High-Pressure Waterflood Installations.” *J. Petrol. Technol.* 23:173 (1971).

Messina, J. P. “Periodic Noise in Circulating Water Pumps Traced to Underwater Vortices at Inlet.” *Power*, September 1971, p. 70.

Mussalli, Y. G., Hofman, P., and Taft, E. P. “Influence of Fish Protection Considerations on the Design of Cooling Water Intakes.” Paper presented at Joint Symposium on Design and Operation of Fluid Machinery, Colorado State University, June, 1978.

Prosser, M. J. “The Hydraulic Design of Pump Sumps and Intakes.” BHRA, July, 1977.

Scotton, L. N., and Anson II, D. T. “Protecting Aquatic Life at Plant Intakes.” *Power*, January 1977, p. 74.

Silvaggio, J. A., Jr. “Air Model Testing to Determine Entrance Flow Fields.” 1st International Pump Users Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, TX, 1984.

Tsai, Y. J. “Sediment Control for a Power Plant Intake Screenwell.” Paper presented at ASME meeting, New York, 1977.

Wallen, P. J., and Towsley, G. S. “A Case History—Improvements to an Existing Cooling Tower Sump and Horizontal Split Case Pumps.” 16th International Pump Users Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, TX, 1999.