SECTION 10.2
INTAKE MODELING

MAHADEVAN PADMANABHAN
JOHANNES LARSEN

PROBLEMS ENCOUNTERED IN PUMP INTAKES

The various hydraulic problems associated with a pump pit include formation of surface and subsurface vortices, prerotation and swirl, and flow separation at or near the suction bell of either wet-pit or dry-pit centrifugal pumps (Figures 1a and 2). Any of these problems can adversely affect pump performance by causing cavitation, vibrations, or loss of efficiency.[1] Usually there is more than a single reason for these problems, and the extent of the combined effects is difficult to predict reliably by mathematical modeling or Computational Fluid Dynamics (CFD). Formation of vortices, for example, even though dependent on suction pipe velocity and submergence, is strongly influenced by added circulation from vorticity sources, such as a nonuniform approach flow resulting from intake and approach channel geometries; rotational wakes shed from obstructions, such as columns or piers; and the velocity gradients resulting from boundary layers at the walls and floor.[2] The circulation contributed by these vorticity sources is unpredictable and strongly dependent on intake design and operating conditions, especially for large pumping units with multiple bays fed by a common approach channel. In these cases, physical modeling is the best way of predicting the behavior of the prototype with a reasonable degree of reliability.

Free Surface Vortices This type of vortex is considered objectionable when it draws air bubbles or an air core into the pump inlet (Figure 1b). Under nominal approach flow conditions, strong air core vortices causing air ingestion with air concentration as high as 10% have been reported.[29] It has been established that an air concentration in the suction pipe of from 3 to 5% can lower pump efficiency.[3] Also, air in the form of large bubbles or slugs can cause the impeller to vibrate. Strong surface vortices that do not draw air are also objectionable if they draw debris into the suction pipes and because their high rotational velocity can reduce the local pressure sufficiently to induce cavitation.

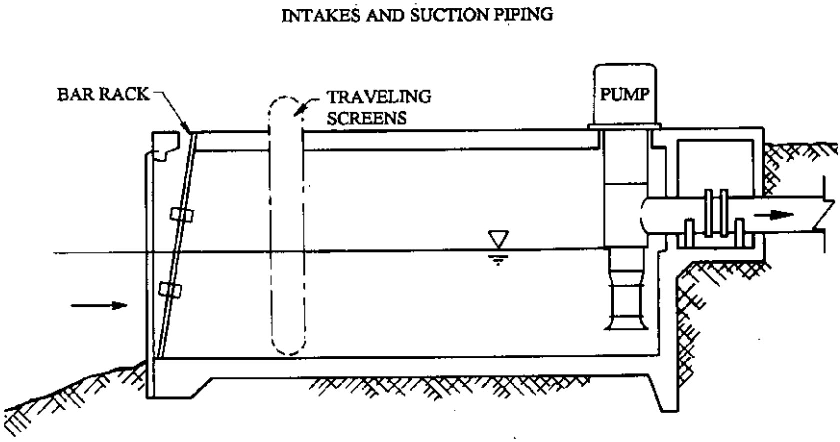

FIGURE 1A Typical vertical wet-pit pump intake.

FIGURE 1B Free surface vortex can form at pump suction bell entrance due to combination of low submergence, high suction velocity, nonuniform approach flow, or other vorticity sources (Courtesy of Alden Research Laboratory, Inc.)

Subsurface Vortices This type of vortex, also known as a submerged vortex (Figures 1c and 2), usually originates from a floor or wall and is induced by vorticity produced in separation zones close to the pump entrance or, in wet-pit pumps, below the bell. The

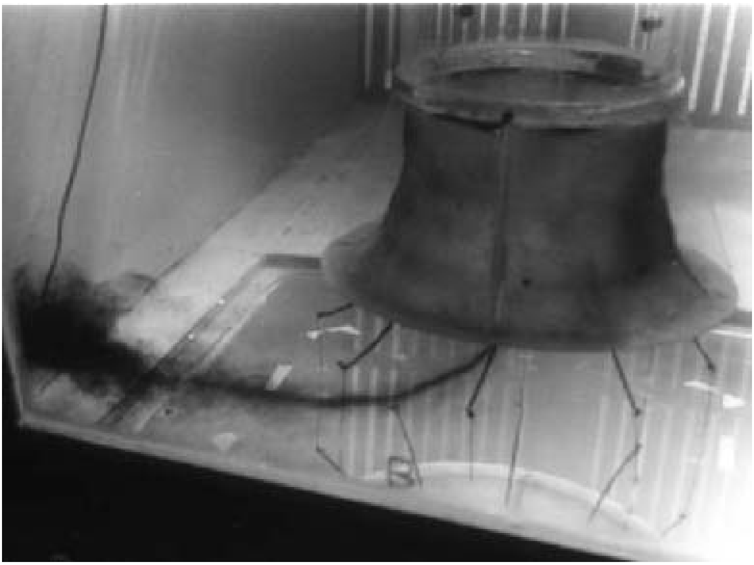

FIGURE 1C Subsurface vortex formed at the side wall due to poor flow guidance to the bell; vortex identified in the model with dye (Courtesy of Alden Resrearch Laboratory, Inc.)

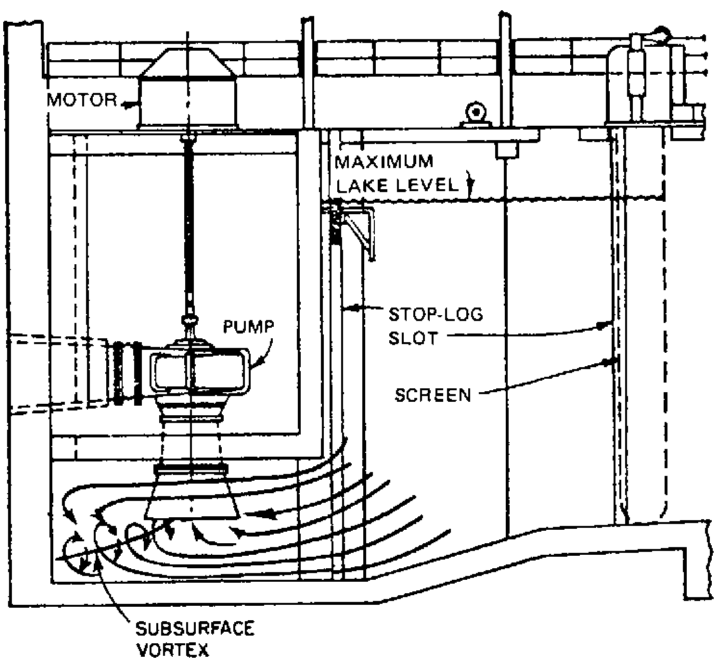

FIGURE 2 Typical vertical dry-pit pump intake. Improper spacing of suction bell relative to floor and back wall caused subsurface vortexing. Model test revealed problem and correction (Reference 4).

reduced pressure in the vortex core causes fluctuating load on the pump impeller along with associated vibration and noise,[4,5] increased possibility of cavitation, higher inlet losses, and decreased pump efficiency, especially when the core pressure is sufficiently low to release dissolved air or other gases from the fluid.

Prerotation and Swirl *Swirl* is a general term for any flow condition (due to vortexing or a pipe bend) where there is a tangential velocity component in addition to a usually predominating axial flow component. *Prerotation* is a specific term to denote a cross-sectional average swirl in the suction line of a pump or, in case of a vertical wet-pit pump, upstream of the impeller.

The prerotation angle θ is a measure of the strength of the tangential velocity component u_t relative to that of the axial velocity component u in the flow approaching the pump impeller; that is, $\theta = \tan^{-1}(u_t/u)$. Adverse effects on the pump are decreased capacity and head when the rotation is in the direction of pump rotation and increased capacity and head when the rotation is opposite the pump rotation (antirotation). The increased capacity is associated with an increase in power requirement and may cause motor overheating.

Prerotation will influence pump performance because the flow approaching the impeller already has a rotational flow field that may oppose or add to the impeller rotation, depending on direction. The design of the pump blades (that is, shape and angle) usually assumes no prerotation, and the existence of prerotation implies flow separation along one side of the impeller blades. The degree of prerotation that should be of concern depends on the type of pump and may not always be known. Prerotation could be quantified in a model by an average cross-sectional swirl angle, determined by detailed velocity measurements, or by readings on a swirl meter. Because swirl decays along a pipe as a result of wall friction, internal fluid shear, and turbulence, the swirl meter in a model suction pipe should be located near the impeller.

Losses Leading to Insufficient NPSH A poorly designed pump intake could result in large inlet losses. Losses caused by screens, poor entrance conditions, vortexing and swirl, and vortex suppression devices may add up to a value so great that the required *NPSH* of the pump is not satisfied. Increased inlet losses due to swirl have been reported in laboratory studies.[6] In a nuclear reactor residual-heat-removal sump model, inlet losses in a preliminary design wherein air core vortices and a high degree of swirl were present were 20% higher than in a revised design with no strong vortices and swirl, and with similar pipe entrance geometry and flows. Because the degree of vorticity and swirl cannot be predicted and it is therefore not possible to calculate inlet losses reliably; they are usually obtained from model studies. With the experimentally derived values of the inlet losses, the *NPSH* available should be checked by recalculation.

DESIGN GUIDES

As a preliminary design guide (Section 10.1), published information may be used to establish pump sump dimensions and a minimum desired submergence.[7–10] Figures 1 and 2 of Section 10.1 show the basic layout of a pump sump and present typical dimensions in relation to the flow required per pump, as published by the Hydraulic Institute.[7] Some of these references may express sump dimensions in terms of bell mouth diameter, and Figure 3 shows the typical relationship between design flow for a given wet-pit pump and the bell mouth diameter required to achieve reasonable and desirable velocities approaching the impeller. One should be aware that the sump dimensions and minimum derived submergences given by design guides are considered applicable for the ideal condition of a simple, straight approach flow with a constant, low approach velocity to the pump sump.

The need for a physical model study still exists when site or operating conditions make the ideal condition impossible. For example, for a particular pump intake configuration, a hydraulic model study indicated that a strong submerged floor vortex existed with floor clearance of 0.5 times the bell diameter (a usual design value) and that a reduced floor

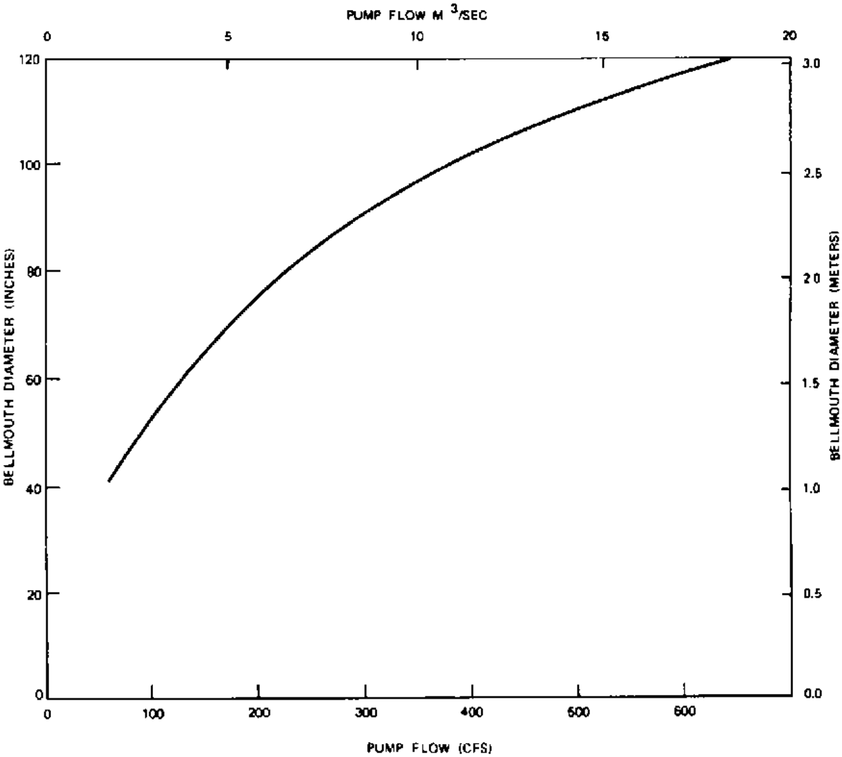

FIGURE 3 An approximate relationship between design flow and required suction bell diameters for vertical propeller pumps for preliminary evaluations.

clearance of 0.3 times the bell diameter eliminated the vortex. Typically, for pump intakes of the following types, model studies should be considered essential:

1. Intakes with nonsymmetric approach flow; for example, an offset in the approach channel
2. Intakes with multiple pump bays with a common approach channel and a variety of pump operating combinations
3. Intakes with pumps of capacities greater than 40,000 gpm (2.5 m^3/s) per pump
4. Intakes with expanding approach channel
5. Intakes with possibilities of screen blockages or obstructions close to suction pipe entrance; for example, reactor containment recirculation sumps, gate guides for dry-pit pumps
6. Dual flow screens intake

For items 1, 2, 4, 5, and 6, a model study is recommended because of the unknown effects a nonuniform approach flow can have on vortexing and swirl. For item 3, considering the cost of large pump installations and the cost of backfit, should problems occur, a model study is recommended to ensure proper pump operation.

MODEL SIMILITUDE

The principle of dynamically similar fluid motion forms the basis for the design and operation of hydraulic models and the interpretation of experimental data. The basic concept of dynamic similarity is that two systems with geometrically similar boundaries have similar flow patterns at corresponding instants of time.[11,12] To achieve this, all individual forces acting on corresponding fluid elements must have the same ratios in the two geometrically similar systems. The condition required for complete similitude may be developed from Newton's second law of motion:

$$F_i = F_p + F_g + F_v + F_t \tag{1}$$

where F_i = inertial force, defined as mass m times acceleration a
F_p = pressure force connected with or resulting from the motion
F_g = gravitational force
F_v = viscous force
F_t = surface tension force

Additional forces, such as fluid compression, magnetic, or Coriolis forces, may be relevant under special circumstances, but generally these forces have little influence and are, therefore, not considered in the following development.

Equation 1 can be made dimensionless by dividing all the terms by F_i. Two systems of different size that are geometrically similar are dynamically similar if both satisfy the same dimensionless form of Equation 1. We may write each of the forces of Equation 1 as

F_p = net pressure difference × area = $\alpha_1 \Delta p L^2$
F_g = specific weight × volume = $\alpha_2 \gamma L^3$
F_v = shear stress × area = $(\alpha_3 \mu \Delta u/\Delta y)$ (area) = $\alpha_3 \mu u L$
F_t = surface tension × length = $\alpha_4 \sigma L$
F_i = density × volume × acceleration = $\alpha_5 \rho L^3 u^2/L = \alpha_5 \rho u^2 L^2$

where

a_1, a_2, and so on = proportionality factors
Δp = net pressure difference
L = representative linear dimension
γ = specific weight = ρg
μ = dynamic viscosity
$\Delta u/Dy$ = depth-wise velocity gradient
u = representative velocity
σ = surface tension
ρ = density
g = acceleration due to gravity

Noting that the kinematic viscosity, ν, is given by μ/ρ, substituting the above terms in Equation 1 and making it dimensionless by dividing by the inertial force, we obtain

$$\frac{\alpha_1}{\alpha_5} E^{-2} + \frac{\alpha_2}{\alpha_5} F^{-2} + \frac{\alpha_3}{\alpha_5} R^{-1} + \frac{\alpha_4}{\alpha_5} W^{-1} = 1 \tag{2}$$

where

$$E = \frac{u}{\sqrt{\Delta p/\rho}} = \text{Euler number} \propto \frac{\text{inertial force}}{\text{pressure force}}$$

$$F = \frac{u}{\sqrt{gL}} = \text{Froude number} \propto \frac{\text{inertial force}}{\text{gravitational force}}$$

$$R = \frac{uL}{\sqrt{\mu/\rho}} = \text{Reynolds number} \propto \frac{\text{inertial force}}{\text{viscous force}}$$

$$W = \frac{u^2L}{\sigma/\rho} = \text{Weber number} \propto \frac{\text{inertial force}}{\text{surface tension force}}$$

Because the proportionality factors α_1, α_2, and so on, are the same in model and prototype, complete dynamic similarity is achieved if the values of each dimensionless group, *E, F, R,* and *W* are equal in model and prototype. In practice, this is difficult to achieve. For example, to have $F_{\text{model}} = F_{\text{prototype}}$ and $R_{\text{model}} = R_{\text{prototype}}$ requires either a 1:1 “model” or a fluid of very low kinematic viscosity in the reduced-scale model. Hence, the accepted approach is to select the predominant force and then design the model according to the appropriate dimensionless group. The influences of the other forces become secondary and are called scale effects.[11,12]

Froude Scaling Pump intake models are generally designed and operated using Froude similarity because the flow is controlled by gravitational and inertial forces. The Froude number is, therefore, made equal in model and prototype:

$$F_r = F_m/F_p = 1 \tag{3}$$

where *m, p,* and *r* denote model, prototype, and ratio between model and prototype.

In modeling a pump intake sump to study the formation of vortices, it is important to select a reasonably large geometric scale to achieve large Reynolds numbers. At a large Reynolds number, energy loss coefficients usually behave asymptotically with Reynolds number. Hence, with $F_r = 1$ and a sufficiently high Reynolds number, the Euler number *E* will be equal in model and prototype. This implies that flow patterns and loss coefficients may be considered similar in model and prototype. From Equation 3, the velocity, flow, and time scales are

$$u_r = L_r^{0.5} \tag{4}$$

$$Q_r = L_r^2 u_r = L_r^{2.5} \tag{5}$$

$$t_r = L_r^{0.5} \tag{6}$$

For example, if a model-to-prototype scale of 1:10 is used, a prototype velocity of 1.0 ft/s (0.3 m/s) becomes a model velocity of $(1 \div 10)^{1/2} = 0.32$ ft/s (0.09 m/s). In the model, all physical dimensions of the pump bays and approach channel are scaled to the ratio 1:10. Submergence, being a linear dimension, is also scaled to 1:10. A flow of 100 gpm (0.006 m^3/s) in the model corresponds to a flow of $100 \times 10^{2.5} = 31{,}623$ gpm (2 m^3/s) in the prototype. Similitude parameters and laws are treated in detail in References 11 and 12.

Similarity of Vortices The fluid motions involving vortex formation in pump sumps have been studied by several investigators. It can be shown by principles of dimensional analysis that the dynamic similarity of fluid motion that could cause vortices at an intake is governed by the following dimensionless parameters:

$$\frac{ud}{\Gamma}, \frac{u}{\sqrt{gd}}, \frac{d}{s}, \frac{ud}{\nu}, \text{ and } \frac{u^2d}{\sigma/\rho}$$

where u = average axial velocity at the bell entrance
Γ = circulation contributing to vortexing
d = diameter of the bell entrance
s = submergence at the bell entrance

ν = kinematic viscosity of water

g = acceleration due to gravity

σ = surface tension of water air interface

ρ = water density

The influence of viscous effects is defined by the parameter $ud/\nu = \mathrm{R}$, the Reynolds number. Surface tension effects are indicated by $u^2 d\rho/\sigma = \mathrm{W}$, the Weber number. As strong air-core type vortices, if present in the model, would have to be eliminated by a modified sump design, the main concern for interpretation of model performance involves the similarity of weaker vortices. If the influence of viscous forces and surface tension on vortexing is negligible, dynamic similarity is obtained by equating the parameters ud/Γ, $u/\sqrt{gd}$, and d/s in model and prototype. A Froude model satisfies this condition, provided the approach flow pattern in the vicinity of the sump, which governs the circulation, Γ, is properly simulated.

Considerable research on scaling free surface and submerged vortices has been conducted in the past few years. From a study of horizontal outlets for a depressed sump, it was determined that for pipe Reynolds numbers above 7×10^4, no scale effect on vortex strength, frequency, or air withdrawal existed.[30] Another study indicated that an inlet Reynolds number of 3×10^4 is sufficient to obtain a good model to prototype correlation of vortices.[6] Surface tension effects on vortexing have been shown to be negligible for Weber numbers greater than 120 based on laboratory experiments.[14]

A review of all available data on model versus prototype vortex intensity indicated negligible scale effects for weak vortices and small scale effects for air drawing vortices, and that this effect could be overcome by a relatively small increase in model flow rate.[27] The model flow rate should only be increased by an amount such that sufficient Reynolds and Weber numbers result. Excessively increasing the model flow, particularly to prototype velocity, produces highly exaggerated vortices incompatible with prototype observations.

DESIGN AND OPERATION OF MODELS

Model Scale Scale effects are less as model size increases but construction and operation costs increase with model size, so a compromise must be made. In general, the formation of vortices, both free surface and submerged, is highly responsive to approach flow patterns, and it is important to select a geometric scale that achieves Reynolds numbers large enough to keep the flow turbulent and to meet the fluid mechanic criteria for minimizing scale effects.[13] Also, one should consider other factors such as access for instruments, accurate flow measurements, and ease of modification in selecting a proper scale.

Information on preferred minimum values of Reynolds number and Weber number, discussed earlier, may be used in designing a model and deciding geometric scale. However, adhering to these limits does not, in itself, guarantee negligible scale effects in a Froude model because these limits are based on tests run under ideal laboratory conditions. In real situations, there is usually more than one source of vorticity generation of unknown extent, and a generalization of scale effects for all cases would be inappropriate. To compensate for such unknown scale effects, a common practice is to test a model at higher-than-Froude scaled flows.

A special test procedure involving high temperatures may be used to determine any scale effects and to project the model results to prototype ranges of Reynolds numbers.[2] The water temperature in the model is varied over a range, say 50 to 120°F (10 to 49°C), and flow velocities in the pipes are varied over a range of values, if possible, up to the prototype velocities. Vortexing and other flow patterns over a range of Reynolds numbers are obtained from these tests and can be used to evaluate any possible scale effects. A prediction of the prototype performance can be made based on these tests.

Extent of Model It is very important to include a sufficient length of approach channel in the model because approach flow nonuniformities contribute greatly to vortex formation and swirl. The decision on what approach length should be included is usually based on the experience and engineering judgment of the model designer. The requirement is essentially the proper simulation of approach velocity profile to the intake and then on to the pump bays. In some cases, considering the cost and time involved in including a sufficient channel length in a pump intake model, a separate, smaller model can be built to determine the approach flow patterns to the intake and these flow patterns then simulated in the pump intake model.[18]

Figure 4a shows a 1:10 geometric scale model of a four-bay pump intake, with each pump designed to draw 140,000 gpm (312 cfs or 8.83 m^3/s) flow. Figure 4b shows a 1:18 geometric scale model pump intake for a flood control project involving several low head high flow pumps—each handling 360,000 to 450,000 gpm (800 to 1,000 cfs) at 12 ft head (22.68 to 28.35 m^3/s at 3.65 m head).

Modeling of Screens and Gratings In addition to providing protection from debris, screens suppress nonuniformities of approach flow. The aspects of flow through screens that are of concern in a model study are (1) energy loss in the fluid passing through the screen, (2) modification of the velocity profile, and (3) production of turbulence. As all these factors could affect vortex formation in a sump with approach flow directed through screens, a proper modeling of screen parameters is important.

The fluid passing through the screen loses energy at a rate proportional to the drop in pressure, and this loss dictates the effectiveness of the screen in altering velocity profiles. The pressure drop across the screen is analogous to the drag induced by a row of cylinders in a flow field and can be expressed in terms of a pressure drop coefficient K (or, alternately, a drag coefficient), defined as[19]

$$K = \Delta p/(0.5\rho u_a^2) = \Delta H/(u_a^2/2g) \tag{8}$$

where Δp = drop in pressure across screen
u_a = mean velocity of approach flow to screen
ΔH = head loss across screen

The loss coefficient is a function of three variables: (1) screen pattern, (2) screen Reynolds number $R_s = u_a d_w/\nu$, where d_w is the wire diameter of the screen, and (3) solidity ratio S', the ratio of closed area to total area of screen (Section 8.1).

If S' and the wire mesh pattern are the same in the model and prototype screens, the corresponding values of K are a function of R_s only. This is analogous to the drag coefficient in a circular cylinder. At values of R_s greater than about 1,000, K becomes practically independent of R_s.[19] However, for models with low approach flow velocity and fine wire screens, it is necessary to ascertain the influence of R_s on K for both model and prototype screens before selecting screens for the model that are to scale changes in velocity distribution.

Velocity modification equations relating the upstream and downstream velocity profiles usually indicate a linear relationship between the two, the shape and solidity ratio of the screen, and the value of K.[20] If wire shapes and solidity ratios are the same in model and prototype, it is possible to select a suitable wire diameter to keep the values of K approximately the same for the model and prototype screens in the corresponding Reynolds number ranges. This produces a head loss across the model screen that is scaled to the geometric scale of the model and that produces identical velocity modifications in model and prototype. Some model designers consider it a conservative approach to leave out the screens in the model altogether, under the assumption that this omission will only worsen the nonuniformity of the approach flow. This approach is not recommended unless it is not practically possible to select an appropriate model screen.

Modeling of Pumps The exterior submerged surfaces of the pump bowl assembly and, for wet-pit pumps, the column including the bell mouth must be modeled to scale as well as the interior geometry from the bell mouth perimeter to the impeller eye. This is to ensure that flow patterns approaching the impeller are properly simulated.

Prerotation induced by the pump's rotating element is discussed in Sections 2.3 and 10.1 and in References 21 to 24. It has been shown that the rotating element does not affect upstream flow patterns when the pump is operated at design flow, and hence,

FIGURE 4A A 1:10 geometric scale model of a four pump-bay pump intake (Courtesy of Alden Research Laboratory, Inc.)

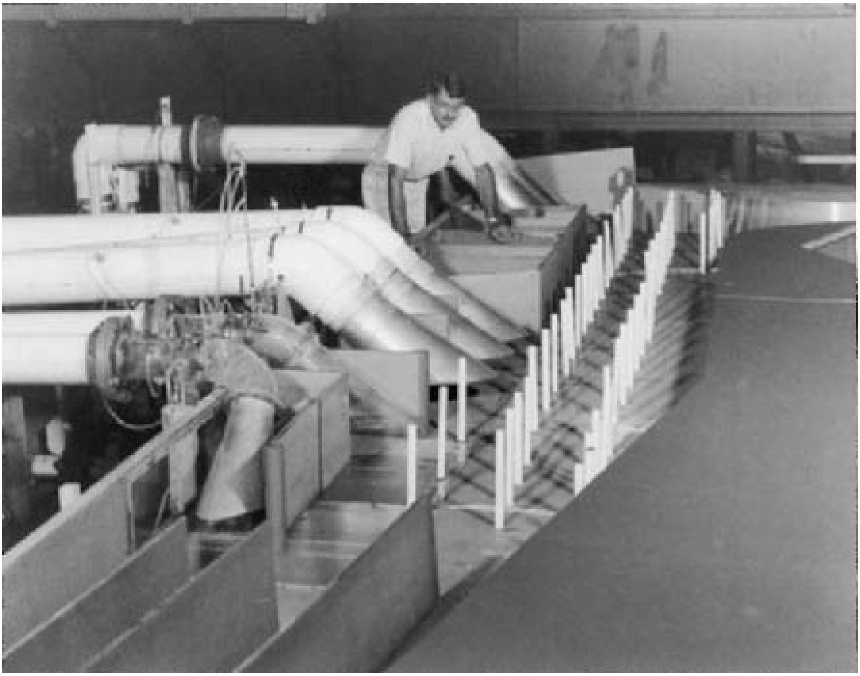

FIGURE 4B A 1:18 geometric scale model of a flood control pumping station with high capacity (800 to 1000 cfs/22.7 to 28.3 m^3/s), low head (about 12ft/3.65 m) pumps (Courtesy of Alden Research Laboratory, Inc.)

including the rotating element in a pump intake study is not necessary. When the pump is operated at less than rated flow, a degree of swirl is induced upstream of the rotating element. This swirl increases rapidly at flow rates less than 45% of rated flow as reversed flow out of the impeller intensifies, and this may affect vortex activity in the pump well and flow distribution to the pump.

Model Operation Operating a model at the prototype suction pipe velocity is thought to be a conservative method to compensate for excessive viscous energy dissipation and the consequent less intense model vortices in a Froude model.[17] This method is often referred to as the Equal Velocity Rule. Operating a model at a higher than Froude scaled velocity should be considered a reasonable procedure for evaluating scale effects. However, operating a model based on the equal velocity rule may not be advisable unless the model is large enough, say at least a 1:4 scale model, because increasing the flow to many times the Froude scaled flow while keeping a scaled submergence could distort the approach flow patterns and turbulent intensities and, thus, cause unrealistic results. In general, a velocity increase to about 1.5 times the Froude scaled velocity can be considered reasonable. More appropriately, the information contained in References 25 and 26 may be used to decide the velocity ratio for exaggeration, which often is considered a function of model scale. If the final recommended design does not show any coherent core vortices in a model, no large scale effects should be expected, and operating the model at higher than Froude scaled flows may be unnecessary in such cases.[27]

Model Cost The cost of model studies varies considerably and is dependent on such factors as number of pump bays, number of operating conditions, and complexity of approach flow. Typically, $40,000 to $90,000 (in 1999 U.S. dollars) can be expected to cover the range from simple one- or two-bay sumps to multibay installations with complex approach flow and several operating modes.

MODEL OBSERVATION TECHNIQUES

Free Surface Vortices Vortices in a model usually contain little energy, and therefore, vortex circulation cannot readily be measured because any measuring device will affect the strength of the vortex. Therefore, vortex strength is best characterized by visual observation and classified by comparison with a vortex strength scale, such as the one shown in Figure 5a. Observations are carried out continuously for a representative period of time, usually one hour prototype, and the resulting data are analyzed in terms of the maximum observed vortex strength and in terms of persistence; that is, the percentage of time a given vortex strength occurred.[28] Vortices of Type 5 and 6 are clearly objectionable in an intake design because of the effect air entrainment and localized pressure reductions have on pump performance. Usually in model studies, a vortex type of 3 is considered the maximum allowable, as prototype strength may be slightly underestimated as a result of possible scale effects.

Sub-Surface Vortices Vortices attached to fixed boundaries below the surface can be difficult to detect and may require a careful search with a dye tracer or an air-injection technique. The vortex strength is best characterized by visual observations and classified by comparison to a vortex strength scale such as one shown in Figure 5b. A strong subsurface vortex may reduce the pressure locally within its core sufficiently low to allow dissolved air to come out of solution resulting in an air core (Type 3 in Figure 5b).

Submerged vortices are most commonly found below the center of wet-pit pumps and around splitter plates introduced to even out flow into the bell mouth. Pressure measurements made with electronic pressure transducers located in the floor under the pump may provide information on rapidly forming and disappearing vortices that cannot be detected with flow visualization. Sub-surface vortices should, as far as practical, be eliminated from any design because their presence imposes fluctuating loads on the impeller and the resulting low-pressure areas may cause local cavitation damage to the pump.

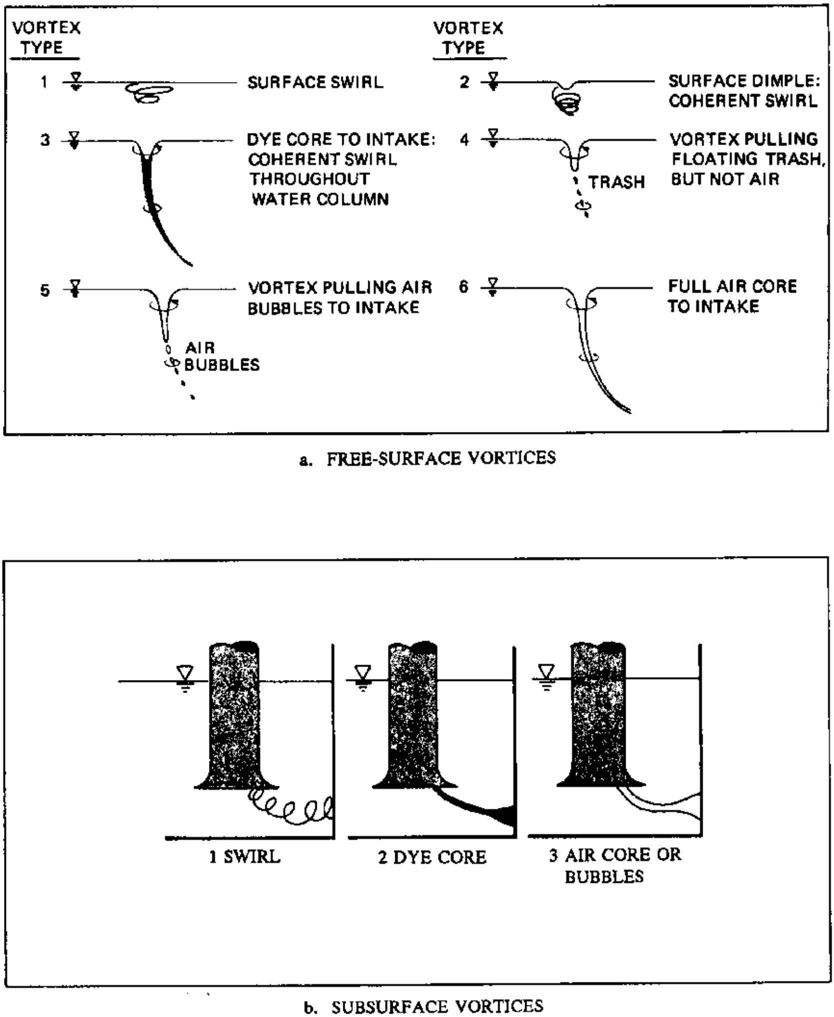

FIGURE 5A and B Vortex strength scale (Courtesy of Alden Research Laboratory, Inc.)

Prerotation The tangential velocity component may be measured by a swirl meter, or vortimeter, which is an impeller with zero pitch in the axial direction mounted axially in the pump column two to four column diameters downstream from the suction bell. A typical vortimeter is shown in Figure 6. Vortimeter rotation is measured by either electronic or manual counting of rotation over a given time. Prerotation can also be determined from velocity traverses obtained with a two-dimensional pitot tube.

Intake Loss Coefficient Water manometers connected to piezometric taps along the pipe are used to measure the hydraulic gradient along the suction pipe. An intake loss coefficient is then determined by extrapolating the measured hydraulic gradient to the suction pipe inlet and computing the head loss $h_L = \Delta h - u^2/2g$, where Δh is the differ-

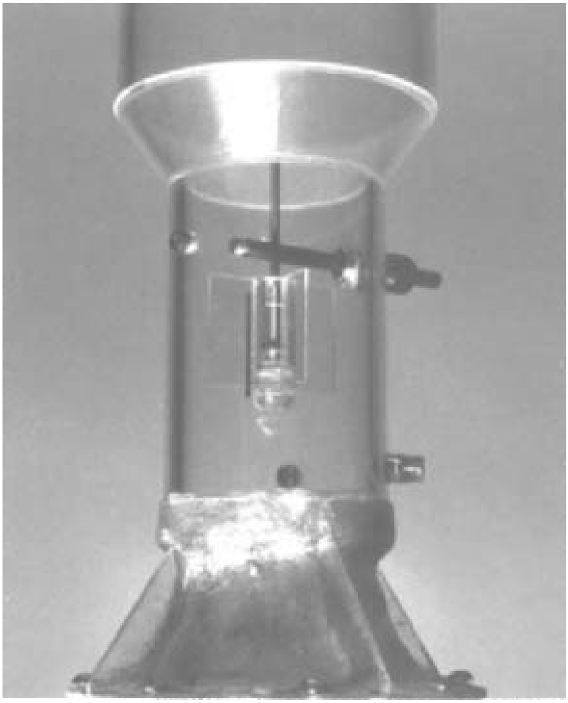

FIGURE 6 Swirl meter in a model pump column (Courtesy of Alden Research Laboratory, Inc.)

ence between the pump pit water surface and the extrapolated pressure gradeline at the pipe inlet. With this procedure, the computed head losses will not include any pipe frictional losses and consequently any model pipe Reynolds number influence is avoided. At the same time, pressure measurements are done far downstream from the region of flow separation near the entrance, and hence, more accurate measurements of total pressure loss, including entrance losses, are obtained. The loss coefficient C_L is usually defined as $C_L = h_L/(u^2/2g)$, where $(u^2/2g)$ represents the velocity head in the pipe. The computed model loss coefficient is valid for the prototype, and hence, the prototype inlet losses are C_L times the prototype pipe velocity.

Flow Distribution Radial flow under the perimeter of the pump bell is required to avoid vibrations and loss in efficiency caused by uneven impeller loading. Flow direction is usually indicated by six to eight yarn streamers equally spaced and mounted on wire supports 0.2 bell mouth diameter below the bell mouth perimeter. Figure 7 shows a typical installation. For an acceptable design, these streamers should not deviate more than 10 to 15° from radial. Velocity distribution approaching the pump is usually measured with miniature propeller meters.

CORRECTIVE MEASURES

Problems in pump sumps such as severe vortexing, intense swirl, or uneven flow distribution at the bell can be reduced or eliminated by deriving proper corrective methods using models. Because no single remedial method to correct these problems exists, it is important to try several suitable methods in the model to derive an effective and practical one.

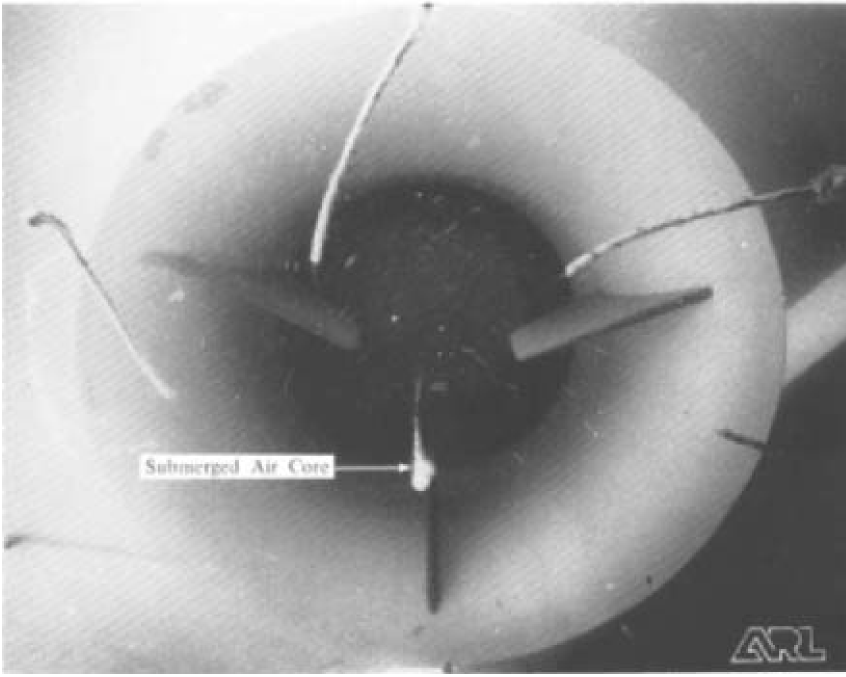

FIGURE 7 Yarn streamers around the suction bell of a model intake to observe flow direction into the bell (Courtesy of Alden Research Laboratory, Inc.)

Even though the pump bell velocity and submergence are very important parameters contributing to problems, it is seldom possible to change these values. Hence, the corrective measures are usually incorporated in the geometry of the pump pit and include the introduction of suitable appurtenant devices.

Severe free surface vortices may be prevented by providing a uniform approach flow to the pumps while maintaining a sufficient submergence. Required submergences can be predicted by empirical equations available in the literature.[31] Relocation of pumps, introduction of horizontal grids below the water surface, changing of wall and floor clearances, improvements in approach channel configurations, and changes in lengths and spacing of piers are some of the common techniques for reducing vortex activity. Surface vortices can also be controlled by installing a curtain wall immediately upstream of the pump. Figure 8 indicates various methods of eliminating poor flow conditions as given in Reference 7.

Submerged vortices are usually eliminated by installing splitter vanes or floor cones under the bell. Variations in floor and wall clearances can also be effective and should be tried in the model. Figure 9 shows a 1:4 scale model of a two-bay pump intake—each pump designed to draw 25,000 gpm (1.575 m^3/s). Flow distributor walls are installed to assure a nearly uniform flow to the pumps, and splitters and fillets are provided to guide the flow into the pump bells without any sub-surface vortices.

The choice of corrective measures should always be made with the pump use in mind. For example, in testing pump intakes that will be used for sewerage, corrections that will allow trash build up, such as vanes or grids, should be avoided. It is also important that a continuous dialogue exist between the intake designer and the test laboratory so the most practical and inexpensive solution for a specific installation can be obtained with a minimum of effort.

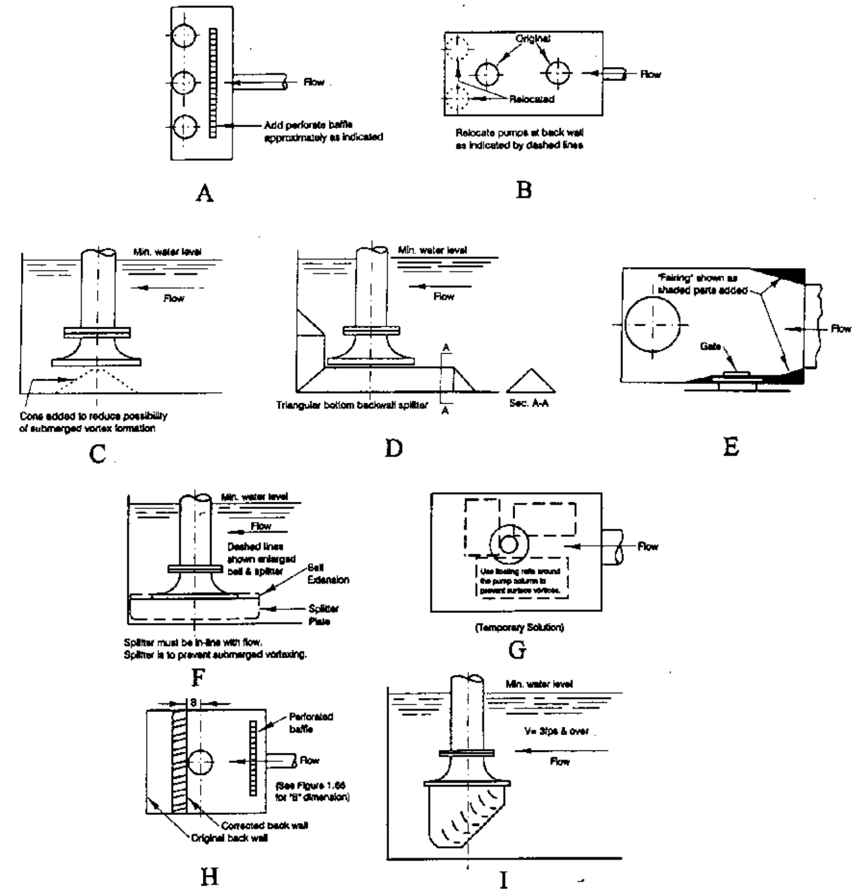

FIGURE 8A through I Common method of eliminating vortexing problems in pump pits (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 7)

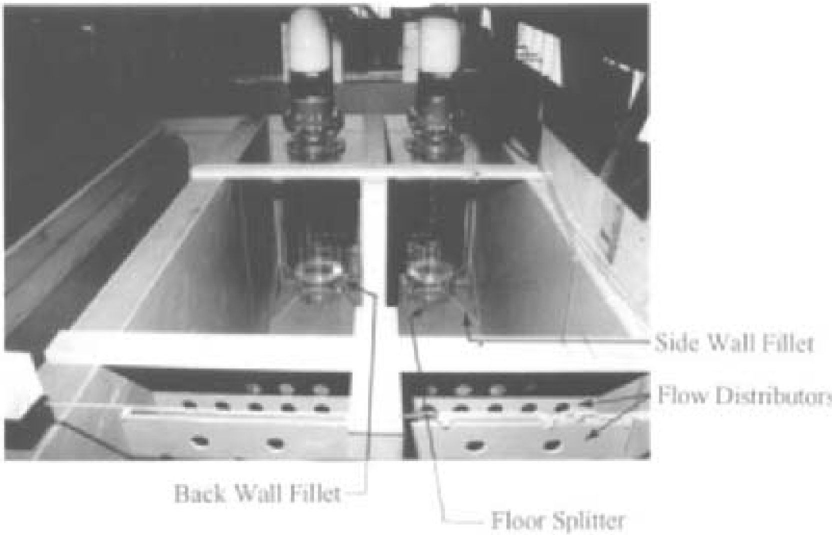

FIGURE 9 A 1:4 geometric model of a two-bay intake showing flow distributors, splitters, and fillets to eliminate objectionable vortices (Courtesy of Alden Research Laboratory, Inc)

REFERENCES

1. Tullis, J. P. "Modeling in Design of Pumping Pits." *Journal Hydraulic Division, ASCE*, September 1979, p. 1053.
2. Durgin, W. W., and Hecker, G. E. "The Modeling of Vortices at Intake Structures." In *Proceedings of the ASCE, IAHR, and ASME Joint Symposium on Design and Operation of Fluid Machinery*, Colorado State University, Fort Collins, CO, June 1978, Vol. 1, p. 381.
3. Murakami, M., and others. "Flow of Entrained Air in Centrifugal Pumps." In *Proceedings of the 13th Congress IAHR*, Kyoto, Japan, 1969, Vol. 2, p. 71.
4. Messina, J. P. "Periodic Noise in Circulating Water Pumps Traced to Underwater Vortices at Inlet." *Power*, September 1971, p. 70.
5. Denny, D. F. "Experimental Study of Air-Entraining Vortices in Pump Sumps." In *Proceedings of the Institution of Mechanical Engineers*, London, 1956, p. 1.
6. Daggett, L. L., and Keulegan, G. H. "Similitude Conditions in Free Surface Vortex Formations." *Journal Hydraulic Division, ASCE*, November 1974, p. 1565.
7. American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Hydraulic Institute, Parsippany, NJ www.pumps.org.
8. Chang, E., and Prosser, M. J. "Intake Design to Prevent Vortex Formation." In *Proceedings of the ASCE, IAHR, and ASME Joint Symposium on Design and Operation of Fluid Machinery*, Colorado State University, Fort Collins, CO, June 1978, Vol. 1, p. 393.
9. Sweeny, C. E., Elder, R. A., and Hay, D. "Pump Sump Design Experience Summary." *Journal Hydraulic Division, ASCE.* March 1982, p. 361.
10. Reddy, Y. R., and Pickford, J. A. "Vortices at Intakes in Conventional Sumps." *Water Power* **24**(3): 108, 1972.
11. Rouse, H. *Handbook of Hydraulics.* Wiley, New York, 1950.
12. Daily, J. W. and Harleman, D. R. F. *Fluid Dynamics.* Addison-Wesley, Reading, MA, 1966.

13. Anwar, H. O. “Prevention of Vortices at Intakes.” *Water Power.* October 1968, p. 393.
14. Jain, A. K., and others. “Vortex Formation at Vertical Pipe Intakes.” *Journal Hydraulic Division, ASCE.* October 1978, p. 1429.
15. Hattersley, R. T. “Hydraulic Design of Pump Intakes.” *Journal Hydraulic Division, ASCE,* March 1965, p. 223.
16. Anwar, H. O., and others. “Similarity of Free Vortex at Horizontal Intake.” *Journal Hydraulics Res,* **2**: 95, 1978.
17. Denny, D. F. and Young, G. A. J. “The Prevention of Vortices and Swirl at Intakes.” Paper No. C1 in *Proceedings of IAHR 7th Congress*, Lisbon, 1957.
18. Larsen, J., and Bozoian, P. M. “Model Studies of Intake Pumphouse for the Perry Nuclear Power Plant Cooling Towers.” Report 153-77/M294EF, Alden Research Laboratory, Inc., Holden, MA, 1977.
19. Padmanabhan, M., and Vigander, S. “Pressure Drop Due to Flow Through Fine Mesh Screens.” *Journal Hydraulic Division, ASCE*, August 1978, p. 1191.
20. Baines, W. D., and Peterson, E. G. “An Investigation of Flow Through Screens.” *Trans. ASME* **73**: 467, 1951.
21. Toyokura, T. “Studies on the Characteristics of Axial Flow Pumps, Part 1.” *Bull. Japanese Society Mechanical Engineering* **4**(14):287, 1961.
22. Minami, S., and others. “Experimental Study on Cavitation in Centrifugal Pump Impellers.” *Bull. Japanese Society Mechanical Engineering* **3**(9): 19, 1960.
23. Kurian, T., and Radhakrishna, H. C. “A Study of the Phenomena of Pre-Rotation at the Suction Side of Centrifugal Pumps.” *Irrigation and Power*, July 1974, p. 381.
24. ”Prerotation in Pumps Induced by the Impeller.” Internal Technical Memo, Alden Research Laboratory, Inc., Holden, MA, July 1980.
25. Zajdlik, M. “New Checking Mode of Model Parameters for Vortex Formation in Pump Tanks.” In *Proceedings of the IAHR 17th Congress*, Baden-Baden, Germany, 1977, p. 379.
26. Chang, E. “Scaling Laws for Air-Entraining Vortices.” Report No. RR1519, British Hydromechanics Research Association, Cranford, Bedford, England, 1979.
27. Hecker, G. E. “Model-Prototype Comparison of Free-Surface Vortices.” *Journal Hydraulic Division, ASCE*, October 1981, p. 1243.
28. Larsen, J., and Pennino, B. J. “Hydraulic Model Study of Jefferson Parish Drainage Pump Station No. 3.” Report 68-81/M429F, Alden Research Laboratory, Inc., Holden, MA, 1981.
29. Padmanabhan, M. “Air Ingestion Due to Free-Surface Vortices.” *Journal Hydraulic Engineering, ASCE*, December 1984, p. 1855.
30. Padmanabhan, M., and Hecker, G. E. “Scale Effects in Pump Sump Models.” *Journal Hydraulic Engineering, ASCE*, November. 1984, p. 1540.
31. “Swirling Flow Problems at Intakes.” Edited by J. Knauss, IAHR Hydraulic Structures Design Manual, A. A. Balkema, Rotterdam, 1987.