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PHYSICAL MODELLING OF FREE SURFACE VORTICES AT PUMP INTAKES - A REVIEW

by

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MANAGEMENT PERSPECTIVE

The prevention of vortices at pump intakes is an important design consideration. Vortices at intake structures cause serious changes in flow patterns, vibrations, objectionable noise, increase energy losses and decrease pumping efficiency. The effective design of pump intakes requires the use of physical scale models to ensure that proper measures are taken to eliminate the formation of vortices. This report presents information which will make it possible to obtain a satisfactory scale model to test different intake design schemes.

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PERSPECTIVE DE GESTION

La prévention des tourbillons à l'entrée des pompes est un facteur de conception important. Ces tourbillons produisent d'importants changements dans l'écoulement, des vibrations, des bruits indésirables, des pertes d'énergie accrues et en rendement de pompage réduit. Pour concevoir des entrées de pompes efficaces, il faut recourir à des modèles physiques à l'échelle pour s'assurer que des mesures adéquates soient prises pour éliminer la formation de tourbillons. Le présent rapport contient de l'information qui permettra d'obtenir un modèle à l'échelle satisfaisant pour mettre à l'essai différents modèles d'entrées de pompe.

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SUMMARY

Using dimensional analysis and data from the literature, criteria have been developed for the design of physical scale models to simulate the formation of vortices at pump intakes. It has been shown that Froudian model-prototype similitude is achieved if the geometry and roughness of the flow boundaries are correctly modelled and the intake pipe Reynolds number is sufficiently large. Available information indicates that models should be built with a scale ratio larger than 1:60. Tests for vortex formation conducted with model intake velocities greater than those dictated by Froudian similitude, should be limited to models of large projects which are required to have small scale ratios. Additional research on the compatibility of model and prototype vortices is required.

RESUME

A partir d'une analyse dimensionnelle et de données publiées, des critères ont été établis pour la conception de modèles physiques à l'échelle permettant de simuler la formation de tourbillons à l'entrée des pompes. On a montré qu'il y a similitude froudienne entre modèles et prototypes lorsque la géométrie et la rugosité des limites d'écoulement sont correctement modélisées et que le nombre de Reynolds dans le tuyau d'entrée est assez élevé. L'information disponible indique que les modèles devraient présenter un rapport d'échelle supérieur à 1/60. Les essais sur la formation de tourbillons menés à des vitesse à l'entrée d'un modèle supérieures à celles dictées par la similitude de Froude devraient être limités aux modèles des grands projets qui doivent présenter de faibles rapports d'échelle. Il faudrait poursuivre la recherche sur la compatibilité des tourbillons des modèles et des prototypes.

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1.0 INTRODUCTION

The potential detrimental effect of free surface vortices on the performance of hydraulic structures and fluid machinery is a major concern for hydraulic engineers. Strong free surface vortices are responsible for flow reductions, vibrations, cavitation, loss in efficiency of pumps and turbines and structural damage (Denny, 1956; Hattersley, 1965; Prossor, 1977; Hecker, 1981). Therefore, considerable effort is made by engineers to develop vortex free designs for intake structures.

The flow patterns associated with free surface vortices are extremely complex. The boundary conditions influencing the flow circulation at a particular intake structure are often site specific and therefore it is not possible to predict analytically whether or not a free surface vortex will occur for a given flow and water level. As a result, model studies are usually conducted to develop a vortex free design or to determine the susceptibility of the prototype to vortex formation. However, considerable confusion and controversy exist as to the importance of scale effects in physical model studies due to apparently conflicting results of past investigations.

In this report, the requirements for model-prototype similitude are examined. Dimensional analysis together with data obtained from the literature are used to establish the basic guidelines for the design and operation of physical models of pump intakes. This work was conducted to provide information for the cost recovery program of the Hydraulics Studies Project at the Hydraulics Laboratory of the National Water Research Institute.

2.0 PRELIMINARY CONSIDERATIONS

2.1 Sources of Vortices

The flow pattern near the intake, which is related to the entry conditions, is the largest single factor contributing to vortex formation (Reddy and Pickford,1974; Quick,1962; Durgin and Hecker, 1978). Vorticity transported towards an intake by the approach flow is intensified near the intake due to the stretching of the vortex filaments. Upstream sources of vorticity, or equivalently, circulation or swirl, determine the supply to the inlet and thus the strength of the free surface vortex. Vorticity sources may be generally categorized into four fundamental types (Durgin and Hecker, 1978), as shown in Figure 1. In the first type, rotation of the fluid near the intake results primarily from the offset in the approach channel. Viscosity does not affect the vorticity flux to the inlet. In the second type, viscosity induced velocity profiles are inherently rotational, with the wall itself a vorticity source (Quick, 1962). Although the approach channel may be symmetric, circulation of one kind or another is usually formed due to small asymmetries in the approach flow patterns. This type of vorticity source is influenced by viscosity and may therefore be Reynolds number dependent. The third source is generated by the tendency of obstructions to have rotational wakes. Typical of this is a pipe or beam which shed alternating vortices. Vortices from one side of the vortex street may continually feed the intake, or an alternating pattern may develop with the intake swirl reversing periodically (Durgin and Heckert, 1978). This third vorticity source can also be expected to be Reynolds number dependent. The fourth source of vorticity is generated by sudden changes in flow boundary alignment. Eddies are shed from sharp corners and this process should also be expected to be Reynolds number dependent (Quick, 1962). If the Reynolds number of the approach flow is low, the eddies will form but will not break away from the corners where they form. However, above a certain Reynolds number, the eddies break away and travel downstream toward the intake.

2.2 Types of Vortices

The type of vortex that will appear at an intake depends on the swirl produced by the approaching flow, the depth of submergence of the intake and the flow rate through the intake. Jain et al (1978) described five stages of development of vortices as shown in Figure 2. For a given inflow configuration and constant discharge, the intensity of the vortex increases as the depth of submergence of the intake decreases. Initially, as the water submergence is gradually reduced, the water surface over the intake will change from a flat surface to a "dimple". As the submergence is further reduced, the "dimple" will gradually change to a vortex, with the core length increasing as the submergence decreases. As the core length increases, a stage is reached when air bubbles are drawn into the intake. Ultimately, the air core will extend fully from the free surface to the intake, thereby allowing continuous passage of air accompanied by the characteristic noise. Careful observations on scale models by Durgin and Hecker (1978) have also shown that the occurrence of a particular type of vortex is dependent on the depth of submergence. A typical relationship between the type of vortex T as classified in Table 1, and the dimensionless core length z/h (z = length of the air core and h = the depth of submergence) is given in Figure 3. As the length of the air core increases when depth of submergence is decreased, as shown in Figure 2 (increasing values of z/h indicate decreasing depth of submergence and vice versa), the curve in Figure 3 shows that for vortices up to Type 4, the type T changes more or less linearly with z/h . In this range the type of vortex occurrence is very sensitive to the depth of submergence. Vortices between type 4 and 6 become progressively less dependent on the depth of submergence as the latter decreases.

The most serious occurrence of vortices are those with a continuous air core. It has been reported by Denny (1956), that for an air entrainment of 1% at the pump intake, pumping efficiency can be reduced by as much as 15%. To avoid such vortices, measures must be taken to prevent the conditions conducive to their formation.

2.3 Need for Model Studies

Theoretical solutions to the formation of vortices at pump intakes are not possible for the complexities of site specific conditions. Sweeney et al (1982) found from a review of seven projects that physical model should be used whenever pump intakes are designed. Guidelines for the design of vortex-free intakes have not yet been clearly established despite decades of research. The more practical of these guidelines gives the required submergence of the intake in terms of the intake diameter (Prosser, 1977), velocity (Denny and Young, 1957), or withdrawal Froude number (Quick, 1970; Reddy and Peckford, 1972; Anwar, 1968). In all these cases the guidelines do not provide any information regarding the expected circulation flow due to approach channel and pump sump conditions, or required apriori knowledge of the flow circulation. In view of the uncertainties in the existing guidelines, it is necessary to develop intake designs using reduced scale hydraulic model studies for each particular case where it is important that no strong vortices occur.

In general, the onset of free surface vortices can be expected to be affected by inertial, gravitational, viscous and surface tension forces as well as the flow boundary configurations. The total dynamic similitude with respect to these forces cannot be obtained in a geometrically similar model using the same fluid as in the prototype. Although the predominating inertial and gravitational forces are reduced similarly in a Froude-scaled model, viscous and surface tension forces cannot be reduced simultaneously in the same proportions. The extra influences of these forces on modelling free surface vortices is called scale effects. Great care must be exercised in accounting for these scale effects when scaling the model results up to prototype values.

3.0 DIMENSIONAL ANALYSIS

The flow into the intake is governed by: flow rate, fluid properties, gravity, intake pipe geometry and the flow approach geometry and properties. Any property, say A, of this flow will be a function of these characteristic parameters. If the flow approach geometry and properties are specified, then the property A can be expressed mathematically by writing

$$A = \Phi\,[Q,\ \rho,\ \mu,\ \sigma,\ g,\ D,\ h_s] \tag{1}$$

where A is a typical dependent variable, Q = the discharge through the intake pipe, ρ = the density of the water, μ = the dynamic viscosity of the fluid, σ = the surface tension of the fluid, g = the acceleration due to gravity, h_s = the depth of submergence at the flow intake, D = the inside diameter of the intake pipe and Φ denotes a function. Taking ρ, D and Q as repeating variables, dimensional considerations yield

$$\pi_A = \Phi_1 \left[\frac{Q}{g^{\frac{1}{2}} D^{5/2}},\ \frac{Q}{\nu D},\ \frac{\rho Q}{\sigma D^3},\ \frac{h_s}{D} \right] \tag{2}$$

where Φ_1 denotes a function, π denotes a dimensionless variable and ν = kinematic viscosity of the fluid. The function of dimensionless variables Φ_1 completely determines the characteristics of the flow such as the circulation, vortex formation, etc. For any particular, site specific critical condition, one may now write

$$\Phi_2 \left[\frac{Q}{g^{\frac{1}{2}} D^{5/2}},\ \frac{Q}{\nu D},\ \frac{\rho Q}{\sigma D^3},\ \frac{h_s}{D} \right]_c = 0 \tag{3}$$

where Φ_2 denotes another function.

In practice, the most often required information is the critical relative submergence of the pipe intake for different withdrawal rates, so that the intake elevations can be properly positioned with respect to the anticipated surface elevations of the approach flow. The critical submergence is generally taken as the lowest submergence at which strong and objectionable vortices do not form. Jain et al (1978) defined critical submergence as the onset of a full air core vortex. Other definition of critical submergence are of course possible. Therefore, equation (3) can be rearranged to give

$$\left(\frac{h_s}{D}\right)_c = \Phi_3 \quad [F, R, W] \tag{4}$$

where Φ_3 denotes a function, $F = Q/g^{\frac{1}{2}}D^{5/2}$, $R = Q/\nu D$ and $W = \rho Q/\sigma D^3$. These terms are a form of the Froude number, Reynolds number and the Weber number, respectively. An alternate but equivalent form of equation (4) can be written as

$$\left(\frac{h_s}{D}\right)_c = \Phi_4 \ [F, E, W] \tag{5}$$

where Φ_4 is another function and $E = R/F = g^{\frac{1}{2}}D^{3/2}/\nu$. The parameter E has the advantage of being unaffected by changes in the discharge Q and is therefore ideal for separating the independent effects of viscosity and surface tension.

As stated in the previous section, total similitude cannot be satisfied if the same fluid is used in the model as in the prototype. Therefore, the functional dependence given in equations (4) and (5) cannot be obtained. Froude number similitude is quite easy to achieve because one needs only to scale down the discharge. However, similitude with Weber number and Reynolds number, which account for the effects of surface tension and viscosity respectively, is rather difficult to obtain. In the next section available experimental results are examined to determined the conditions under which the Weber number and the Reynolds number can be ignored.

3.1 Surface Tension

The most revealing and conclusive experimental results are those obtained by Jain et al (1978) using water and water-isoamyl alcohol solution to study the effect of the surface tension on the formation of air entraining vortices, while keeping viscosity constant. Tests were conducted in a circular tank in which the circulation, normally generated by the approach flow, was generated by means of an assembly of adjustable guide vanes, placed concentric with the vertical pipe intake. The vanes could be set at any desired angle with respect to the radial direction. Typical test results are shown in Figure 4 for three different angles of vane settings. For each fluid the critical submergence is given by a smooth curve for the three vane settings of 20, 45 and 60 degrees. The curves for the two fluids are virtually coincident for each vane setting, indicating that for the data shown, there is no significant effect of the surface tension on the depth of submergence for air entraining vortices.

The plots in Figure 4 are essentially equivalent to the solution of a special case of equation (5). Since the tests were conducted for a fixed pipe diameter and constant viscosity, then E is constant for all the test shown in Figure 4. Therefore Figure 4 is the same as a plot of $(h_s/D)_c$ versus F with W as a parameter for the particular value of E. Since the Weber number varies with the discharge through the pipe, then a different value of W is obtained for each value of Q for a given fluid. Therefore, the agreement between two curves for fluids having significantly different surface tensions for different induced swirl at the intake, shows conclusively that surface tension is not important and that the Weber number can be omitted from Equation (5). Analysis of additional test results by Jain et al (1978) gave similar results and it was concluded that the surface tension does not affect the formation of vortices.

Dagget and Keulegan (1974) conducted tests with mixtures of water and glycerol and found that there was no significant effect of surface tension on the vortex flow. Dhillon et al (1981) after a literature review concluded that surface tension can be omitted in the modelling of the vortex phenomenon. In contrast to this, Yildrin and Jain (1981), in a numerical study, found that the effect of surface tension on the free surface vortex profile became significant at low values of circulation. However, it is probably an acceptable practice to neglect surface tension effects when modelling swirls and surface dimples (Hecker, 1981). Odgaard (1986) determined with the aid of mathematical modelling that surface tension effects are not significant as long as $W > 720$, which is easily achieved in a physical model.

In view of these findings, for flows of practical importance, the Weber number may be omitted from equation (5), which results in

$$\left(\frac{h_s}{D}\right)_c = \Phi_5 \; [F, R] \qquad (6)$$

and

$$\left(\frac{h_s}{D}\right)_c = \Phi_6 \; [F, E] \qquad (7)$$

where Φ_5 and Φ_6 denote the respective functions.

3.2 Effect of Viscosity

Tests were conducted by Jain et al (1978) using a chemical substance called Cepol as an additive to the water to increase its viscosity. The experimental setup was the same as that used for determination of the effect of surface tension. Typical test results are shown in Figure 5 for two different vane settings each representing a different induced flow circulation. The plot in Figure 6 shows the variation of critical submergence as a function of the intake velocity for liquids of different kinematic viscosities namely, water and the cepol solution. The smooth curves fitted to the data show that an increase in the kinematic viscosity, from water to cepol solution, decreases the critical submergence as well as its rate of change with change in pipe flow velocity. Jain et al (1978) attribute this to the reduction in the strength of circulation in the region of the vortex core when the liquid has a higher viscosity. As the increase in viscosity means lowering of the Reynolds number, this indicates that vortices are less likely to form in models than in prototypes. Therefore great care must be taken to ensure that models are built large enough so that scale effects as a result of viscous forces are not significant. Clearly, the range of flow conditions over which the Reynolds number is important must be established.

To examine the implication of the effect of viscosity for physical modelling Jain et al (1978) plotted $(h_s/D)_c$ versus F with E as a parameter for a fixed intensity of swirl. Plotted data together with fitted curves are given in Figure 6. The curves show that the dimensionless critical submergence is a function of both the Froude number and E for the particular geometry of the experimental apparatus. The general equation for the curves in Figure 6 can be written as

$$(h_s/D)_c = \alpha F^{\beta} \qquad (8)$$

where α and β are experimental coefficients. Jain et al (1978) found that β = 0.5 for all tests conducted, whereas α was itself a function of E for a given approach flow condition which can be expressed in general as

$$\alpha = f\ [\ E\ ] \qquad (9)$$

where f = a function. Curves according to equation (9) are given in Figure 7 as α versus E for different levels of swirl used in the experiments by Jain et al (1978) (ie: different alignments of concentric flow guiding vanes). The curves clearly show that regardless of the conditioning of the approach flow, the effect of viscosity becomes insignificant when $E > 4 \times 10^4$.

The results from Figure 6 and 7 were used to convert E to corresponding values of the Reynolds number, say R, which is more familiar to most engineers and is often more practical for scaling of model roughness and other properties which are affected by the viscous forces and are usually documented in terms of the Reynolds number. It was seen in Figure 7, that viscous effects are negligible when $E > 4 \times 10^4$ for all flow conditions. Since E = R/F, the lowest permissable Reynolds number, say R*, can be determined from the relationship

$$R^* \geq 4 \times 10^4 F \qquad (10)$$

Equation (10) states that the critical Reynolds number increases as the Froude number increases. When F = 1.0, $R^* > 4 \times 10^4$ and when F = 10.0, $R^* > 4 \times 10^5$. Dagget and Keulegan (1974) found that $R^* > 2.5 \times 10^4$. Anwar (1968) and Anwar et al al (1978) expressed Reynolds number as $R_r = Q/\nu h_s$ and proposed that $R_r > 3 \times 10^3$. Prossor (1977) suggests that $R_r > 2 \times 10^4$. It is interesting to note that R_r increases as h_s decreases. This tends to support the observation by investigators that greater scale effects are encountered when air core vortices are modelled. Nevertheless, considering the results of Jain et al (1978) as being most conclusive, values of R* should be in accordance with equation (10).

4.0 DESIGN AND OPERATION OF PHYSICAL MODELS

4.1 Scaling Relationships

The basic relationships for site specific model-prototype similitude are given by equation (6) and (7). It has been stated by Quick (1962), Hattersley (1965), Quick (1970), Dagget and Keulegan (1974), Sharp (1981) and shown by the data in Figures 6 and 7, that a scale model of a pipe intake should be built in accordance with Froude number similitude. In using these Froude models one must ensure that scale effect due to viscous forces can be neglected. In other words, the geometric scale ratio, say λ_y (λ denotes scale ratio of the indicated variable), is chosen so that $E_m > 4 \times 10^4$ or $R_m > R*$ (subscript "m" refers to model). Considering the first condition, one must have

$$\frac{g^{\frac{1}{2}} D_m^{3/2}}{\nu} > 4 \times 10^4 \tag{11}$$

where D_m = the diameter of the model intake pipe. Given that $\gamma = 10^6$ m^2/s and g = 9.81 m/s^2, the minimum permissible size of the model intake pipe is D_m = 6 cm. Based on this requirement, the model scale λ_y can be chosen and the remaining parameters can be determined from Froude scaling. The above requirement has been derived from the results of Jain et al (1978) in which critical submergence was defined as the condition for the occurrence of full air core vortices. As this type of vortex is most susceptible to viscous effects, adhering to the above requirement should ensure that viscous scale effects are negligible for other vortex conditions as well.

The requirements for a Froudian model are expressed by the relationships

$$\lambda\pi_1 = 1 \; ; \; \lambda_F = 1 \qquad (12)$$

in which $\pi_1 = (h/D)_c$. The required scaling relationships can then be obtained by writing

$$\lambda\pi_1 = 1 : \lambda_{hs} = \lambda_D = \lambda_y \qquad (13)$$

and

$$\lambda_F = 1 : \lambda_Q = \lambda_D{}^{5/2} = \lambda_y{}^{5/2} \qquad (14)$$

For economic and spatial reasons it is best to design a model as small as possible, while at the same time maintaining dynamic similitude. The smallest possible model is dictated by equation (11). Although these factors are basic to proper similitude, other factors such as measurement accuracy and smallest size of structural components must also be considered. In addition, the minimum similitude criteria should be considered to be only approximate and therefore one may wish to build a model larger than indicated by the minimum similitude criteria alone, to include a margin of safety. Therefore, it is often necessary to use a scale factor m ($m = 1/\lambda_y$) which is larger than dictated by equation (11). This and other aspects of the operation of physical models of pipe intakes will be discussed next.

4.2 Operation of Physical Models

The essence of modelling is to build a model as small as possible while at the same time maintaining satisfactory similitude. This means a value of m as large as possible. Prossor (1977) has shown that models of pump sumps and intakes should not have values of m larger than 30. Anwar (1968) suggests that for satisfactory simulation of prototype performance, a Froude model reproducing total circulation or curved flow patterns, should have $m < 20$. Hecker (1981) reports that good results have been obtained for models of pumped storage reservoir intakes with $m = 100$. Clearly, depending on the approach channel conditions and modelling objectives, the choice of m for physical models of intake structures and their forebays is very important.

There has been much controversy regarding the proper operation of the scale models. It has been shown that models should be operated according to the Froudian laws of similitude. However, many investigators have modelled vortex phenomena using higher intake pipe velocities than those required with a Froudian model in an effort to obtain additional assurance that no viscous scale effects in modelling air core vortices occur. Rohan (1966) suggests an increase in model velocities by a factor of R_p/R_m (m and p denote model and prototype respectively). Denny and Young (1957) concluded from model tests that under similar submergence conditions but using prototype velocities, the model behaves the same as the prototype. Springer and Patterson (1969) investigated vortex formation experimentally and found that for higher outlet velocities and increasing swirl, a vortex forms at greater submergence, thereby indicating that a minimum velocity exists below which a vortex will not form at all. Iverson (1953), Dicmas (1967), Zagdlik (1977), Dicmas (1978) and Chang (1979) have shown that satisfactory agreement between model and prototype is obtained by operating the model at flows larger than dictated by Froude scaling.

In an attempt to resolve the difference of opinion in the operation of scale models, Dhillon et al (1981) modelled a prototype intake structure using four differrent criteria. Hecker (1981), conducted a survey of 22 projects for which model studies were conducted using (a) Froudian similitude, (b) velocities larger than those required for Froude scaling, but less than protype velocities and (c) model velocities equal to prototype velocities. The information as abstracted from Dhillon et al (1981) and Hecker (1981) is broken down into modes of model operation and model-prototype agreement and this is given in Table 2. In all cases the values of R or E were greater than R* and 4 X 10^4 respectively. The results are examined with special emphasis on the scale factor m to determine the most promising course of action for the operation of physical models of pump intakes.

4.2.1 Froude scale models ($\lambda_F = 1$)

It is apparent from Table 2 that out of a total of 27 projects, 22 or about 82% were operated at the Froude scale. These are summarized in Table 3. Out of these 22 models 16, or 73% had good agreement between model and prototype performance. It is particularly noteworthy that 11 of the 16 successful tests concern good comparison of negligible or weak vortices. This is a significant finding because negligible scale effect for weak swirls or dimples is implied. The remaining 6 cases concern good comparison of air core vortices. These cases are not as instructive because any scale effects would dictate that an air core model vortex certainly represents an air core prototype vortex (Hecker, 1981). The greatest success occurred when the scaling factor was less than 60 as shown in Table 3. Indeed, 17 out of 22 models were designed for values of m in the range 10 < m < 60, with an overall success rate of 77%. The total number of models with m > 60 is 5, which represents about 23% of models designed with λ_F = 1. Out of this total of 5 models, 3 or 60% were successful.

Some of the models shown in Table 2 were built with values of m < 60, even though criterion given by equation (11) permit larger values of m. A case in point is a model examined by Hecker (1981) having a scale ratio of 1:50. For this case, $R = 3.7 \times 10^7$ and $F = 0.25$, and, according to equation (10), the minimum allowable Reynolds number $R^* = 10^4$. Using this value of R^* for R_m, one obtains

$$\frac{R_p}{R_m} = \left[\frac{D_p}{D_m}\right]^{3/2} = m^{3/2} = \frac{3.7 \times 10^7}{10^4} \qquad (15)$$

and $m = 239$. Therefore, the criterion $E > 4 \times 10^4$ actually allows a much larger value of m than the one which was used. However, other factors not documented in the original references for Table 2, such as boundary roughness and alignment, may have required a larger model in order to avoid scale effects in the approach flow. The Reynolds number for the approach flow must be sufficiently large so that proper generation of vorticity at the flow boundaries and discontinuities, if the latter exist, can occur. This may partly explain why the success rate for prediction of prototype behaviour using Froude models appears to decrease as the value of m increases. Therefore, if there are any uncertainties regarding approach flow conditions it is probably most prudent to build model pump intakes as large as possible.

It can also be seen from Table 2, that 6 of the models under predicted the formation of vortices in the prototype. The scale factors for these six models were in the range $24 < m < 122$, which is quite similar to that for the models given in Table 3. For these cases, the difference in model performance does not appear to be due to the selection of scale ratio. Hecker (1981) suggests that the deficit in model vortex production was due to scale effects, even though the basic criterion for R^* had been satisfied. All 6 projects had occasional air core vortices and therefore some scale effects in Froude models of air vortices is indicated.

4.2.2 Models with $\lambda_F > 1$

The six cases in Table 2 which show that prototype vortices were stronger than those observed in the model when the model was operated at Froude scale flows, suggest that there was some scale effect that was not experienced with the sixteen successful models operated in a similar manner. One possible explanation for this discrepancy is that the Reynolds numbers for the approach flow may not have been sufficiently large to avoid significant viscous scale effects in the generation of vorticity near the intake. Other possibilities are site specific factors, which are not readily definable or may have been unknowingly omitted in the model construction. Therefore, even though the criterion of $E > 4 \times 10^4$ is satisfied, the model may still produce insufficient vortex intensity due to indirect viscous scale effects in the approach flow. As a result, although the limits of vortex formation at the intake itself are reasonably well defined, the uncertainty in the overall model performance have led researchers to operate Froude scaled models at flows greater than those dictated by Froudian similitude.

The main purpose of modelling is to determine with an acceptable degree of certainty the type of vortex formation, or the lack thereof, that is going to occur in the prototype. Vortices can occur in various stages of intensity as shown in Table 1. However this classification of vortices pertains only to the flow conditions in the near vicinity of the pipe intake. Even though the criterion of $E > 4 \times 10^4$ has been satisfied, viscous effects may still be significant as a result of the conditions of the approach flow. Therefore, if unknown viscous effects are active in the model, then vortex intensity can be expected to be less than in the prototype. In order to overcome viscous scale effects, investigators operate the model at velocities greater than required by Froudian similitude. Proper interpretation of the effect of increasing the model velocity, requires some means of comparing model and prototype vortices. Durgin and Hecker (1981) suggest to identify the vortices according to type as shown in Table 1. Since each type represents a given range of swirl intensity, then for a given model geometry, the vortex type can be expressed in functional form as

$$T_n = q\ [\ F,\ R\ ] \qquad (16)$$

where T_n = the nth vortex type with n varying from 1 to 6 in accordance with the classification in Table 1 and q denotes a function. Projection of model vortex intensity to the prototype implies that T_n must be determined by increasing the Reynolds number without changing the Froude number. Two ways of accomplishing this are to change the water viscosity by changing the temperature or to build models of different scale ratios. Clearly, the most practical of these is the first case, however, Durgin and Hecker (1981) found that this method has provided inconsistent results and that operation of the model at velocities above those indicated by Froude scaling was necessary. Such velocities imply that $\lambda_F > 1$.

The condition $\lambda_F > 1$ for a given model can be expreesed in general terms by writing

$$\lambda_v = k\ \lambda_y^{1/2} \qquad (17)$$

where k = a coefficient > 1. When k = 1, $\lambda_F = 1$ and the model is operated according to Froudian similitude. From equation (17), one can write

$$V_m = \frac{k}{m^{1/2}}\ V_p \qquad (18)$$

where V_m and V_p are the model and prototype pipe flow velocity respectively. It is clear from equation (18) that the upper limit of the coefficient k is m , because for this condition $V_m = V_p$. Therefore, the model can be operated at increased flows for $\lambda_r > 1$ in the range

$$1 < k \leq m^{\frac{1}{2}} \tag{19}$$

Clearly, the operating range of the model depends on the scale ratio m and thus, the geometry of the approach flow. Excessive distortion of the approach flow will result in distorted swirl at the intake thereby providing misleasing results (Gulliver et al, 1987).

The test summary in Table 2 shows that 4 models were operated at flows for values of $2 < k < 4.5$. Out of this total, 3 models gave good agreement between model and prototype and one model exhibited greater vortex activity than was observed in the prototype. The models were built over a wide range of scale ratio. The two largest models gave good results even though values of k may have been as high as 4.5. In contrast, the two models with the smallest scale ratio (ie: largest m) were less successful. This again shows that the size of the model should be chosen with great care and particular attention must be paid to the details pertaining to the approach flow conditions.

One model test in Table 2 is for the case of $k = m^{\frac{1}{2}}$ (ie: $V_m = V_p$). Results reported by Dhillon et al (1981), indicated that the model vortices were considerably stronger than those observed in the prototype. Similar observation were made in an earlier study by Linford (1965). Denny and Young (1957), on the basis of one observation of a prototype pump previously modelled at a 1:16 scale ratio, concluded that using prototype velocities in models of such a scale ratio may be proper. However, as demonstrated by Dhillon et al (1981), the use of the equal intake velocity concept would seriously undermine the primary Froude scale criterion used to achieve proper approach flow patterns and resultant circulation at the intake. Hecker (1981) suggests that models operated at flows larger than those dictated by Froude scaling should be restricted to models of large projects for which scale ratios are small ($m > 100$). In such models, velocities can be increased sufficiently with only small increases in flow through the model.

5.0 CONCLUSIONS

5.1 Using dimensional analysis and data from the literature it was shown that for a given geometry of an intake facility, the depth of intake submergence at which a vortex will form (critical submergence) is dependent on the Froude number and Reynolds number. The effect of the Weber number was found to be insignificant for the size of scale models required for satisfactory Froudian similitude.

5.2 The critical submergence depends on the type of vortex being studied. For a given intake and discharge through the pipe, the smallest critical submergence occurs for air core vortices. These vortices are the most difficult to model successfully, because they are most subject to viscous scale effects in the model.

5.3 It was found that, if boundary geometry and roughness is correctly modelled, then for satisfactory Froudian similitude it is required that $E > 4 \times 10^4$ which means that the intake pipe Reynolds number must be greater than $4 \times 10^4 F$. This is equivalent to a minimum model intake pipe diameter of 6 cm when water is used as the fluid. However, even when the $E > 4 \times 10^4$ criterion is satisfied, one may still get viscous scale effects because the rest of the model, in particular the approach flow, may not be free of scale effects. The $E > 4 \times 10^4$ criterion applies to the flow in the immediate vicinity of the intake.

5.4 Scale effects due to viscosity are strongly dependent on the geometry and the roughness of the flow boundaries in the approach flow. Therefore it is absolutely necessary that the model scales are large enough to ensure that model surfaces in contact with the flow properly represent prototype roughness.

5.5 Careful attention must be given to scaling the approach topography and boundary roughness so that the vorticity flux approaching the intake is correctly modelled. Model tests should include a careful delineation of various vortex types. Notation should be made of vortex location and persistence.

5.6 Most successful Froude scale models were found to have scale ratios greater than 1:60 ($m < 60$). The maximum value of m that can be used must be small enough to avoid scale effects in the approach to the intake pipe.

5.7 The concept of increasing model flows to values greater than dictated by Froude scaling, should be limited to models with small scale ratios (ie: models of very large prototypes) for which small flow increases yield velocities up to prototype values. Large increases in flows are not justified because they distort the approach flow pattern.

5.8 Vortices with intensity ranging from surface swirl to dye core vortex are not subject to significant scale effects in Froude scale models. In contrast to this, model air core vortices are more susceptible to scale effects, especially in models with large values of m.

5.9 A dye core from the free surface to the inlet (Type 3 in Table 1) is a reasonable demarcation between acceptable and unacceptable vortex conditions since this represents the initiation of coherent subsurface swirl. Because of scale effects in some Froude scale model (ie: large values of m), it is possible that a dye core vortex in the model may be indicative of an air core vortex in the prototype. Research should be conducted to examine this possibility.

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TABLE 1

CLASSIFICATION OF VORTICES AT INTAKES

(Durgin and Hecker, 1978)

VORTEX TYPE, T	DESCRIPTION
1	Coherent surface swirl
2	Surface depression
3	Coherent dye core
4	Suction of slightly buoyant particles
5	Air bubbles pulled into intake
6	full air core to intake

TABLE 2

NUMBER OF PROJECTS WITH MODEL - PROTOTYPE COMPARISONS

Categories	$\lambda_F = 1$	$2 < \lambda_F < 4.5$	$V_m = V_p$
(a) Prototype vortices stronger or more persistent than in model	6	0	1
(b) Model and prototype vortices essentially equal	16	3	0
(c) Prototype vortices weaker or less frequent than in model	0	1	1

TABLE 3

EFFECT OF SCALE FACTOR ON MODELLING AT FROUDE SCALE

Range of m	No. of Models	No. of Successes
10 < m < 30	7	5
31 < m < 60	10	8
61 < m < 80	1	0
81 < m < 100	1	1
101 < m < 150	3	2
Total Models	22	16

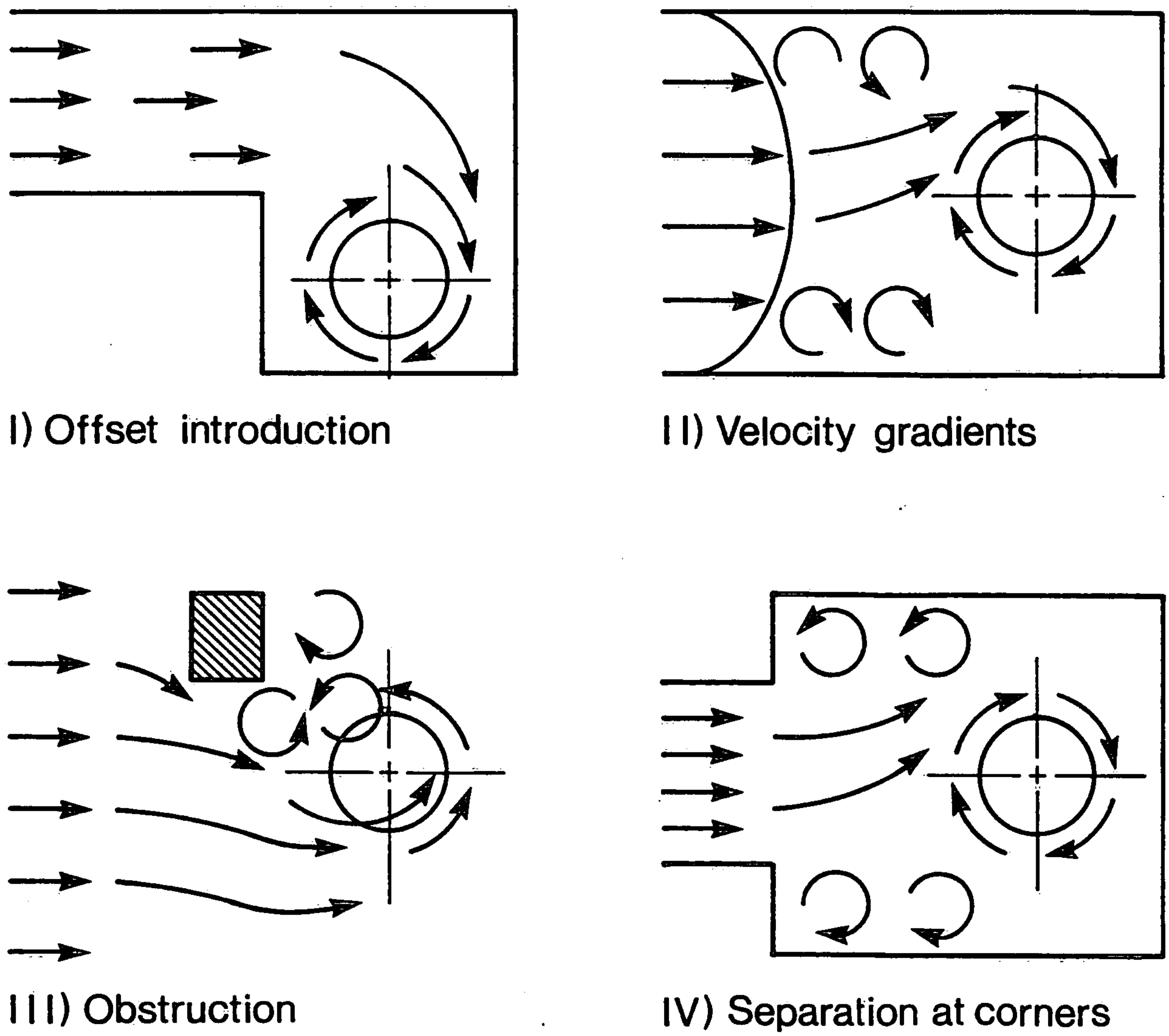

Figure 1 Potential sources of vortices at intake structures.

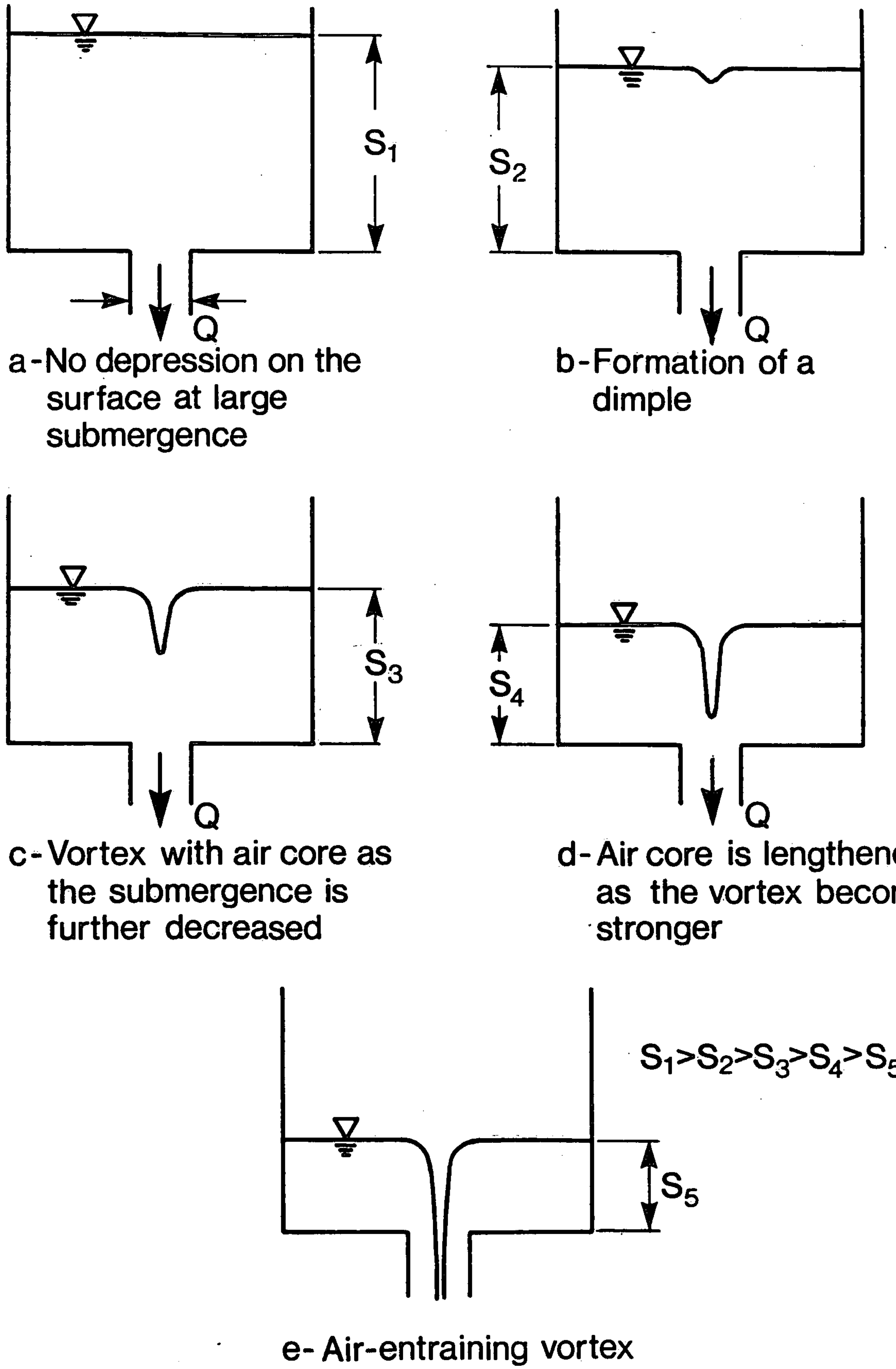

Figure 2 Stages of vortex development for constant discharge Q (Jain et al, 1978)

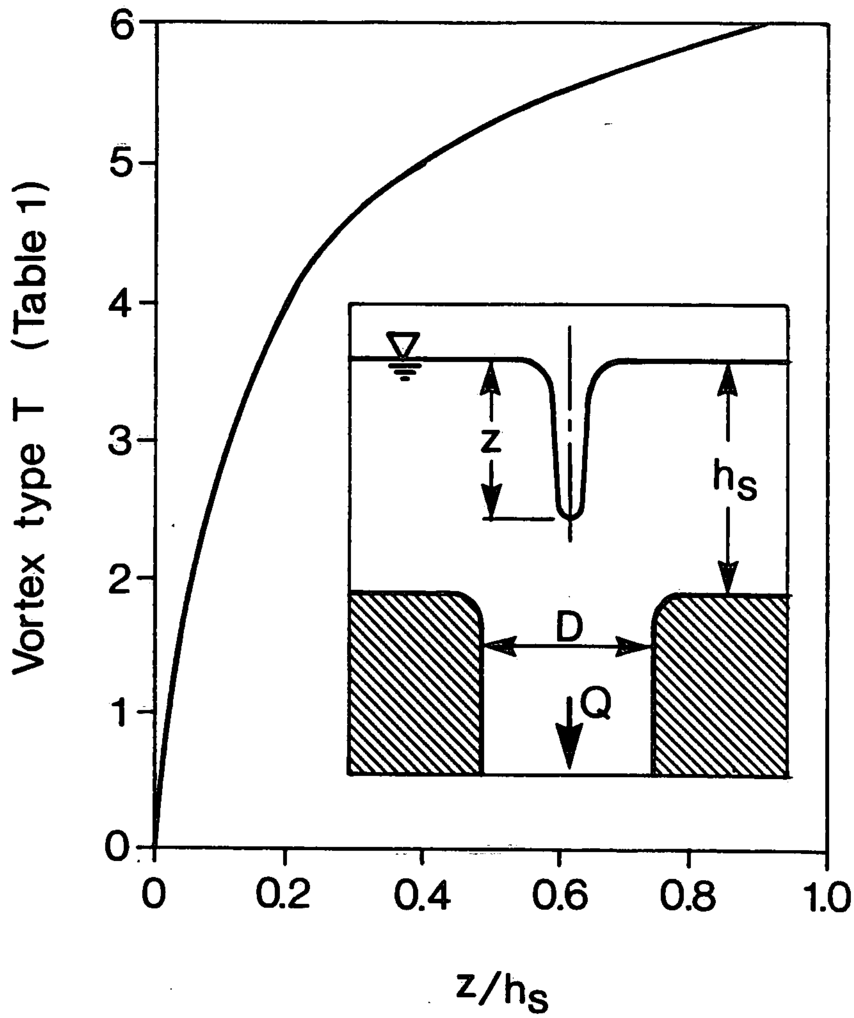

Figure 3 Vortex type versus core length.
(Durgin and Hecker, 1978)

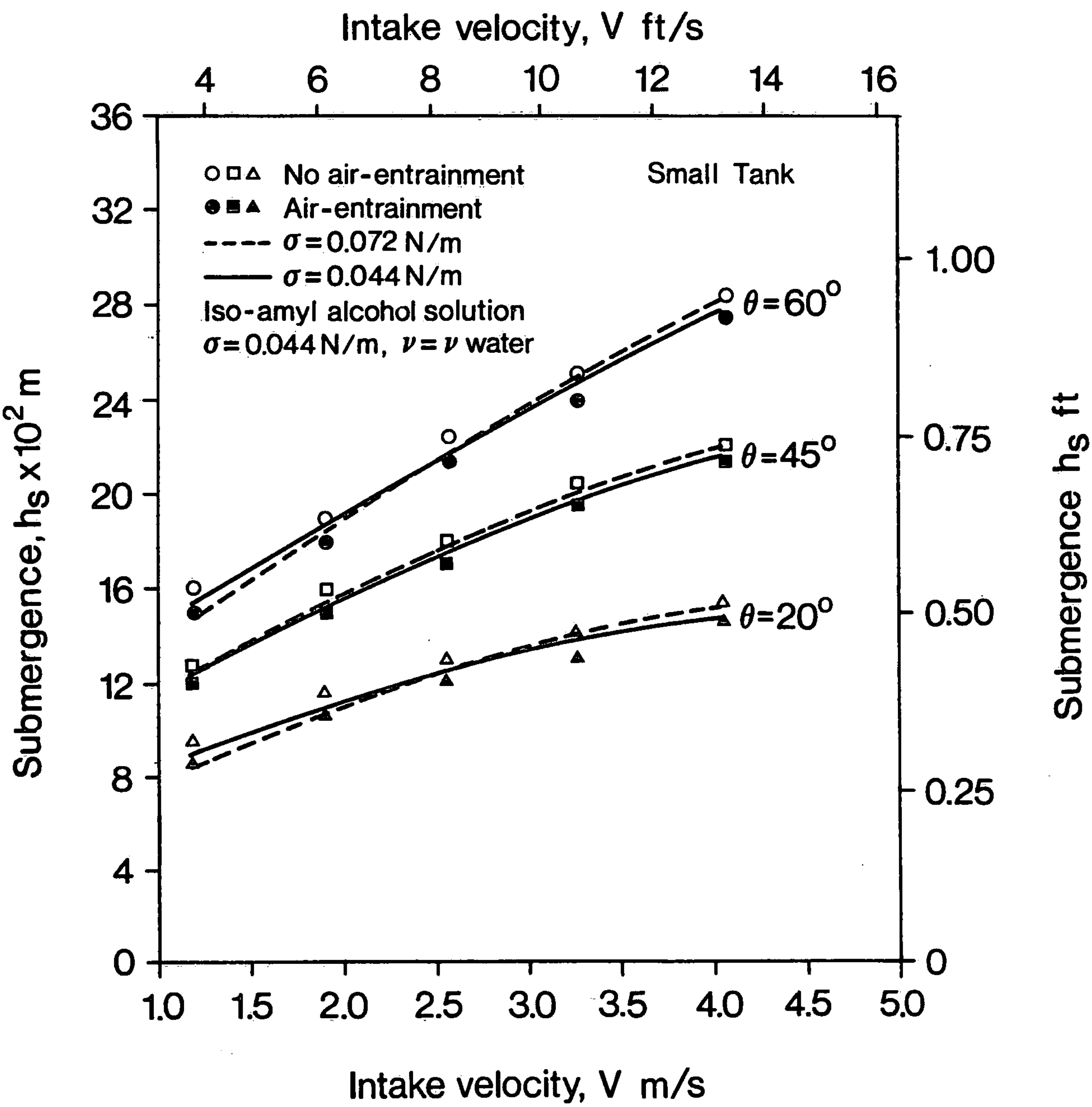

Figure 4 Effect of surface tension on vortex formation at constant viscosity. (Jain et al, 1978)

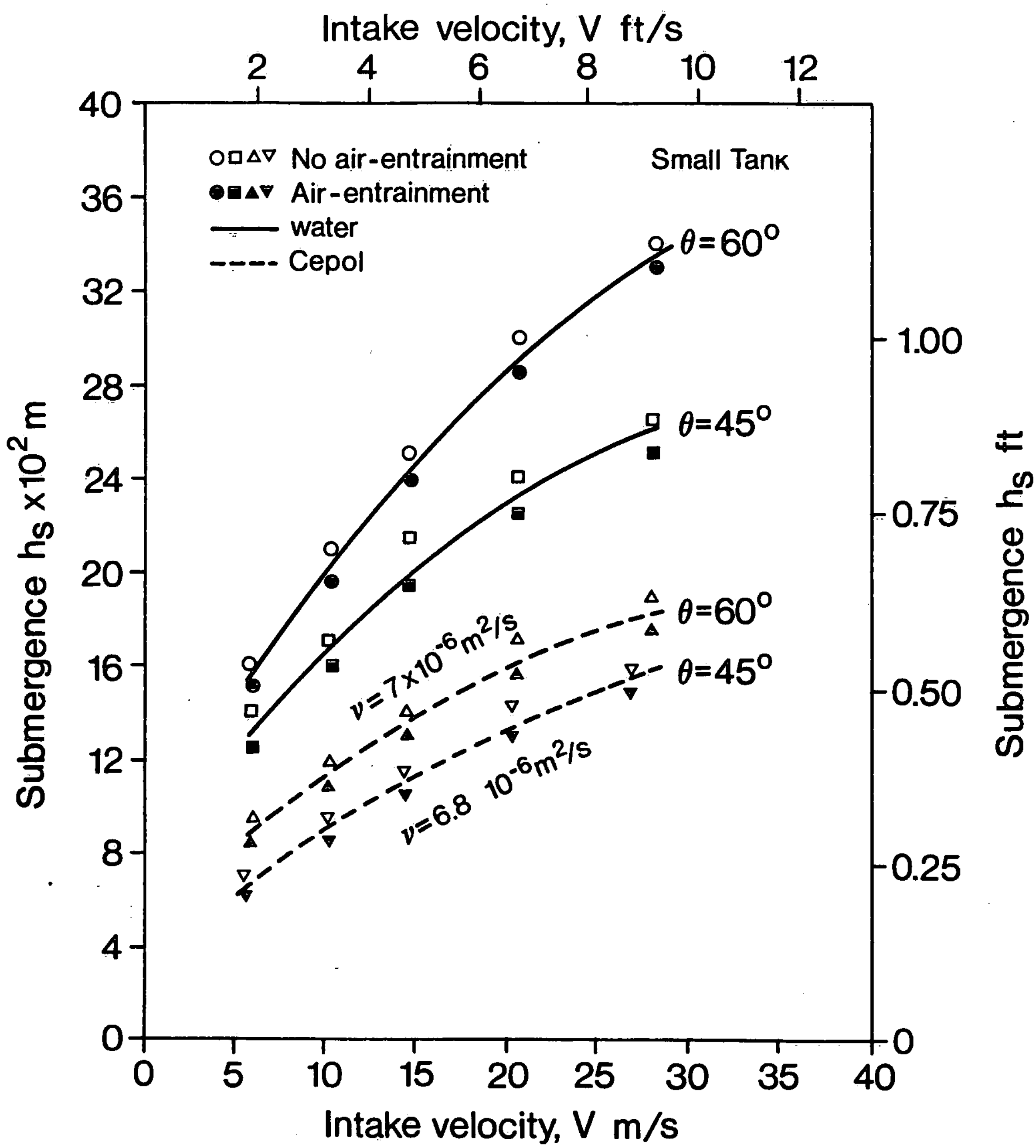

Figure 5 Effects of viscosity on vortex formation (Jain et al, 1978)

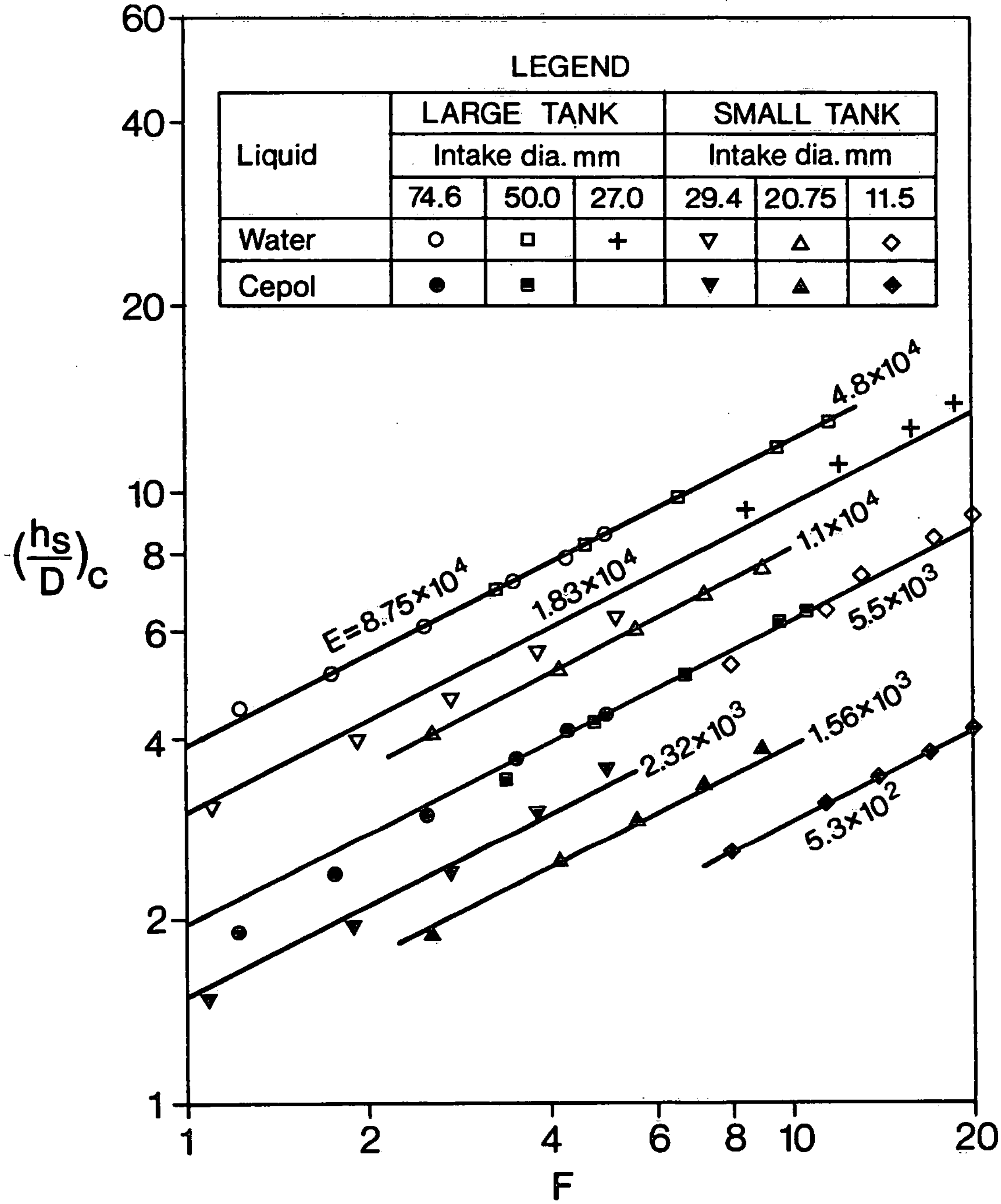

Figure 6 $(h_s/D)_c$ vs F with E as a parameter
(Jain et al, 1978)

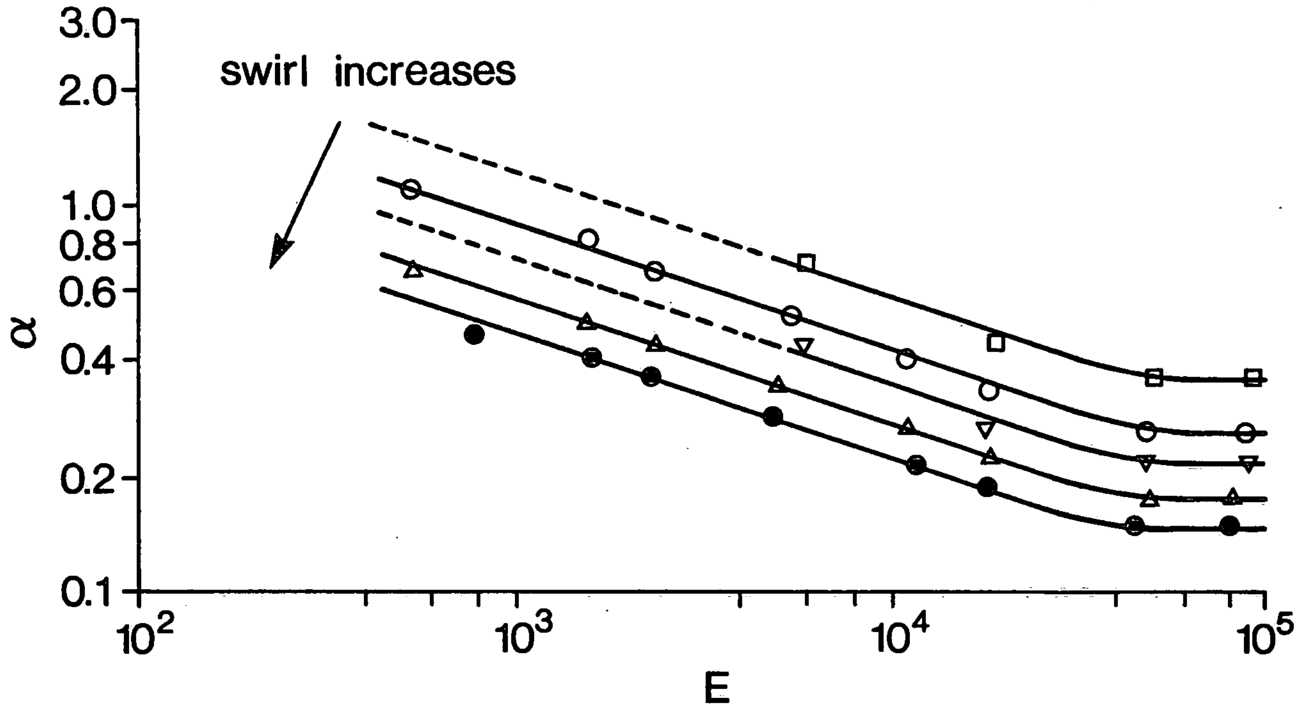

Figure 7 α as a function of E for different intensities of swirl. (Jain et al, 1978)