A STUDY OF THE FORMATION OF VORTICES
WITHIN A MULTIPLE PUMP SUMP

by

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ABSTRACT

A literature survey was undertaken to collect information on multiple pump-sump design, the problems involved and solutions offered by various authors. This led to a study of the vortex forming mechanisms and a detailed examination of their significance to the design of a particular multiple pump-sump.

An 'in-line' sump layout was selected because it was considered that such an arrangement may give the smallest sump volume, which often proves the cheapest. Though this layout is known to give rise to severe vortex formations, a better understanding of the vortex forming mechanisms might permit the development of a satisfactorily operating 'in-line' sump.

Hot film anemometry techniques were used to monitor the turbulence levels within the sump and provide an index of the condition of the flow entering the bellmouth intakes. Considerable practical difficulties were experienced in making the anemometer probe function reliably. Because of the inherent inaccuracy of any unidirectional sensor within a fluid, the normal turbulence intensity parameter $\frac{\overline{u'}}{\overline{U}}$ was replaced by measurements of the main stream velocity fluctuations (dU), which offered a better representation of the comparative state of the flows within the sump.

It was found that the introduction of various flow guides about the bellmouth intakes to physically suppress eddy shedding from the adjacent upstream pumps, significantly improved the sump performance. The tapering
of the sump by varying the angle between the channel walls to reduce the deceleration of the flow along the channel, also added to the flow stability.

It was concluded that the present study indicated the feasibility of constructing a workable 'in-line' sump configuration, and that further investigations incorporating direct pump efficiency tests in conjunction with turbulence measurements would be worth while.
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INTRODUCTION

Vortex formations may be found in many real fluid flow situations, and can vary in magnitude from the huge tornado vortex encountered in weather systems, to the small drain vortex found in the household bath or sink. In many engineering applications it is the flow past totally or partially immersed bodies and flow at water intakes that are of prime interest; and it is with the latter that this study is principally concerned.

Circulation or swirl near a pump intake may well lead to non-uniform pre-rotation of the water within the suction pipe and a consequent drop in pump efficiency.

Pump design starts from two basic premises, namely that there is uniform flow across the bellmouth and that the flow approaching the impellor throat is axial. Any departure from either or both of these conditions may lead to non-uniform loading of the impellor blades and an alteration in the angle of attack between the approaching fluid and the leading edge of the impellor blades, resulting in a consequential impairing of performance in efficiency. In extreme cases there may even be sufficient swirl in a poorly designed sump to cause stall, resulting in unsteady flow conditions at the leading edges of the impellor blades and causing vibration and noise within the pump. This is particularly noticeable in the case of high specific speed axial and mixed flow machines, where the impellor is usually designed assuming a free vortex distribution, as opposed to the forced vortex concept used in centrifugal or low specific speed pumps. (c.f. Appendix I)
Furthermore, the swirl itself may be unstable, and in severe cases can lead to a fully developed vortex with a hollow core, through which air may be drawn into the bellmouth intake reducing the pumped volume and efficiency. The amounts of air entering a machine may be as high as 10-15% by volumetric ratio*, 1% by volume of free air may reduce the efficiency of a pump by as much as 5-15%,* and many of the medium and lower specific speed units, operating under suction lift conditions, will deprime with less than 10% by volume of air.

Air-entrainment may arise in three ways. Firstly, as already mentioned, through air-entraining vortices; secondly by the induction of bubbles already in suspension in the approach flow and thirdly by the release of dissolved air into cavities formed in the water by separation of the stream lines. However, the decrease in pump delivery, brought about by the presence of constant air-entraining vortices, has a tendency to reduce the severity of the vortex itself, which further reduces the air-entrainment. Consequently the delivery readjusts, increasing slightly and reinducing the conditions for air-entrainment. Thus, it is possible for the pump delivery to fluctuate slowly between wide limits, and for the vortex to continually change in severity.

Should the initial severity of the vortex be such that the air-entrainment itself be unsteady, then again, the result would induce vibrations detrimental to the mechanical functioning of the machine.


In a case where the fluid itself has corrosive tendencies, such as associated with salt water cooling systems frequently used in power plants, the ingress of air tends to act as a corrosive catalyst, to the detriment of the system.

There are three fundamental approaches to the examination of the vortex phenomena: the purely theoretical analysis; empirical design based upon real practical experience; and the approach based on the conceptual understanding of the vortex forming mechanisms, gained from controlled laboratory testing of models and which enable the problem to be designed out.

The absence of precise mathematical theories from which to predict the formation of vortices, has led to an emphasis upon the latter two. The empirical approach, though useful for the normal standard size and types of sump design, offers little more than a general outline guide. Most present day trends towards the larger hydraulic projects are for pre-constructional model-testing, though even these precautions do not always ensure complete similarity between model and prototype because of the inherent impossibility of attaining complete Dynamic Similarity.

This thesis reviews both the past and present methods of vortex investigation, and reports the results of experiments aimed at understanding and classifying vortex forming mechanisms, so that the problem in pump sumps may be logically overcome.
CHAPTER I

SYNOPSIS OF A LITERATURE SURVEY

SECTION I: GENERAL DESCRIPTION OF THE VORTEX

Vortices in the past have often loosely been categorized into three distinctive types: the 'local' vortex, the 'column' vortex, the 'drain' vortex. All three are free vortices with the same basic mechanisms of formation.

Owing to the presence of circulation, concentrated or localized swirl and boundary layer momentum exchange mechanisms (c.f. Appendix II), the incoming flow will take a rotation about an axis within the field from which the intake draws surface water. Angular momentum will be conserved within a free vortex, and the swirl velocity will increase towards the centre resulting in ever increasing rates of fluid shear, until at some finite radius solid body rotation will occur, which is the region defined as the vortex core. The point of transition ultimately depends upon the ratio of the inertia to viscous forces. Initially the vortex starts off as a small surface depression, which may gradually develop into a full spiral vortex, with an air core (c.f. Fig. I).

The depth to which the initial surface depression extends depends upon:

a) the strength of the vortex; (defined as the product of the whirl velocity \(V_t\) and the radius \(r\) at any given point i.e. \(V_t \cdot r = \text{constant}\). (\(V_t\))
THE DEVELOPMENT OF AN AIR-ENTRAINING VORTEX CORE.

General direction of fluid flow: →

a) → e) Represent 5 approximate stages of vortex development.

a) & b) Gravitational forces predominant.

c) & d) Surface tension transition & predominance.

e) Inertia forces predominant.
b) the downstream components of flow in the vicinity of the core, taken towards the drain or intake.

THE LOCAL VORTEX: Often encountered in pump sumps, and occurs when highly organized vorticity within the incoming flow forms at the surface and is drawn down into the intake (Fig. I) giving the impression that the initial core has been stretched. It is temporal with respect to space and time, and is usually caused by boundary layer discontinuities which lead to separation, insufficient submergence, local swirl and even air-entrainment.

THE COLUMN OR CONCENTRIC VORTEX: Occurs under severe conditions of high swirl and/or low submergence of vertical intakes, causing the water surface to be drawn down around the periphery of the intake pipe concentrically, and may well be so severe that air-entrainment occurs. Such air entrainment is invariably very noisy, and will set up conditions unstable enough to cause rough running and even depriming of the pump.

THE DRAIN VORTEX: Arises when flow exiting through a bottom outlet under gravity develops a central spiral air-case. Such conditions may be found in constructions such as bellmouth spillways, vertical overflow pipes, submerged hydraulic turbine intakes and container tank bottom exits, all of which experience a reduction in discharge when the air-core develops.
It is not difficult to see that the above categorizations are essentially one and the same mechanism. That the 'local vortex' is simply an inverted extension of the 'drain vortex', and that the 'concentric vortex' is a severe form of the 'local vortex' resulting from general circulation around the sump, rather than from a more localized swirl.

Comparison and Constituents of Open and Closed Vortex Cores

A free irrotational vortex is one whose individual fluid elements have no spin as they rotate about the central axis, and comply with the theoretical relationship:

\[ V_t \cdot r = \text{(constant)} \]

Each stream tube has a different path length and a different tangential velocity, indicating relative motion or 'slip' between, but not about, the individual stream lines. (c.f. Fig. 2a). The magnitude of the shear force losses due to this relative streamline motion is very small and so Bernoulli's equation may be applied at different sections of the vortex, assuming a constant energy distribution throughout the region. Hence, as the pressure increases with increase in radius across the vortex, the velocity decreases, in accordance with Bernoulli. (c.f. Fig. 2c).

\[ \frac{V_{t1}^2}{2g} + \frac{P_1}{\rho} = \frac{V_{t2}^2}{2g} + \frac{P_2}{\rho} \]

where

\[ r_1 > r_2 \quad \text{and} \quad p_1 > p_2 \]

\[ z_1 > z_2 \quad \text{and} \quad V_{t1} < V_{t2} \]
THE PRESSURE AND DISTRIBUTIONS WITHIN:
A FREE VORTEX AND A FORCED VORTEX.

\[ \omega = 0 \]
Particles have no spin

\[ \omega \neq 0 \]
Particles have spin

\[ V_t \cdot r = \text{constant} \]

\[ V_t / r = \text{constant} \]

\[ \frac{P - P_0}{\rho} = \frac{K^2}{2} \left[ \frac{a^2 - r^2}{a^2 r^2} \right] \]

Energy = constant
Vorticity = zero
Circulation = constant

Energy not constant
Vorticity = constant
Circulation = constant
This holds until the velocity gradient between consecutive stream-lines becomes large enough to render significant viscous shear losses. At this point there is a change in the type of flow pattern. Either an air core will form if the velocity is high enough to cause the pressure at the surface of the fluid to fall to a value equal to atmospheric, or solid body rotation of the fluid will commence, forming a forced vortex fluid core separated from the main free vortex body by undefined region. (c.f. Fig. 3).

Because the energy distribution across a forced vortex is no longer constant, Bernoulli's equation may now only be applied along individual stream-lines and not at corresponding points on adjacent ones. The pressure and velocity distribution also are shown in Fig. 3, which illustrates that a free spiral air-entraining vortex in the initial stages of its development, before a full air core has been formed, is a combination of both an open and closed vortex.
PRESSURE AND VELOCITY DISTRIBUTION WITHIN A FREE SPIRAL VORTEX WITH A PARTIALLY DEVELOPED AIR CORE.

Fig. 3

Velocity distribution

Section xx

Section yy

Velocity distribution

$V_t = \frac{K}{r}$

$tangential velocity$

$V_1$ - at a point

$r$ - radius at a point

$P$ - pressure at a point

$P_0$ - atmosphere pressure

$K$ - constant

$\rho$ - density
SECTION II: THE FORMATION OF VORTICES WITH RELATION TO THEIR PHYSICAL SURROUNDINGS

In the past, there have been many conceptions concerning the mechanisms by which vorticity is generated. Fig. 4 illustrates one such set of vortex mechanisms, indicating that concentrated vortices arose from several independent mechanisms, namely: separation and boundary, shear layers, and a third rather vague category referred to as swirl or circulation.* These mechanisms were seen to be dependant upon both the physical geometry and the existing state of flow.

It is now more generally accepted that the boundary layer exchange of diffuse vorticity and the shedding of highly organized discreet vortices from discontinuous boundary layers are the two basic methods by which vortices are organized. Hence by careful symmetrical design, with considerations for elementary stream-line flow, mass circulation may be designed out. But, the method by which diffuse vorticity and localized swirl are generated from the boundary layer still remain a significant problem. It would seem logical therefore, to divide any sump analysis into three sections:

A) THE APPROACH CHANNEL is normally considered to be the culvert connecting the water source (river, lake or ocean, etc.) to the bellmouth intake region (i.e. the sump). It frequently happens that the intake is situated at the watersource and hence the system will have no approach channel.

*(Not in the strict definition of circulation.*)
PREVIOUSLY HELD CONCEPTIONS OF THE
MECHANISMS OF VORTEX GENERATION.

BOUNDARY LAYER GENERATION

(CIRCULATION)
(from asymmetrical boundaries)

(WITHOUT IRREGULARITIES) (WITH IRREGULARITIES)

SEPARATION

MOMENTUM EXCHANGE

DIFUSE VORTICITY

(LOCAL VORTEX) (CONCENTRIC VORTEX) (DRAIN VORTEX)
When there is an approach channel, however, it should be long enough to establish a bi-stable boundary layer, encouraging the balanced and uniform stream-line distribution condition required for the flow entering either the sump or the intake itself. Also, to avoid separation of the flow, the channel wall design should be devoid of sudden changes/discontinuities in its surface geometry. To encourage overall stability, the main stream flow velocity should be kept high enough to overcome the local velocity fluctuations by reducing the channel's cross-sectional area correspondingly.

These local velocity fluctuations may be brought about by any of a number of causes: large scale separation; boundary layer irregularities; sudden changes within the boundary layer continuity, such as sudden expansions or contractions; alternate eddy shedding from submerged objects permanently or temporarily caught in the main stream flow, and even wind disturbances upon the free water surface.

B) THE SUMP: Experience has shown that a uniform and axial flow condition is desired at the bellmouth intake and that the largest single factor governing vortex formation is the flow pattern within the sump, which is in turn dependant upon the flow conditions at entry to the sump itself.

As in the case of the approach channel, the boundary layer's controlling influence upon the flow within the sump remains paramount provided that the mean channel velocity exceeds a critical minimum; also the vorticity generated within the boundary layer is a function of the main stream velocity. (c.f. Appendix II). The subsequent
result of these features were conveniently described by Hattersley(7) using a general swirl parameter $R\sqrt{f}$, who observed that swirl formation seemed to occur in approximately three phases (c.f. Appendix III).

Fig. 5 represents a typical swirl development at a side pit. It was seen that providing the sump through flow velocity* is great enough to keep the $R\sqrt{f}$ parameter within the second linear phase, swirl generation will be kept to a minimum. Further, it was suggested(10) that the three categories of vortices (Section I) corresponded approximately with these phases of swirl formation which is debatable and is discussed further in Section VI.

C) THE SUCTION INTAKE: To conform with pump impellor design criteria, a uniform and axial flow is required at the eye of the impellor.

To avoid separation and non-uniformity of the flow occurring at the transition from the sump to pipe, a bellmouth piece is used to accelerate the flow in the right direction and proportions. The flaring of the bellmouth helps to lower the initial bellmouth entry velocity, which in turn has a tendency to discourage vortex-core entrainment.

*The channel velocity passing and going on down stream of the intake.
THE 3 PHASES OF SWIRL AFTER HATTERSLY (7) (for a typical side channel sump).

Fig. 5

\[ \tan \alpha = \frac{7T}{N d_p} \]

Air entrainment imminent

Irregular & Random swirl formation

Linear decrease of \( \tan \alpha \) with increase in \( \text{RE} \sqrt{T} \)

Very severe swirl

\[ \phi = \text{phase} \]
SECTION III: THEORETICAL APPROACHES TAKEN TO THE ANALYSIS OF SPIRAL IRROTATIONAL VORTICES

To date, the basic approach of most theoretical analyses has been the direct combination of the Navier-Stokes equations with the continuity equation, in cylindrical co-ordinate form, followed by the application of the vortex's boundary conditions to give a solution. However, any exacting analysis must consider the condition of constant pressure over the free surface, whose position is initially unknown, and thus the analysis falls into the class of complicated free surface boundary value problems.

To derive any solution, it has always been necessary to make certain simplifying assumptions, and restrict the analysis to a particular type of flow, i.e. assumptions of an incompressible inviscid fluid of laminar motion are generally made. Further, since the Navier-Stokes equations are non-linear partial differential equations, assumptions as to whether the velocity fluctuations are to be considered small or large, steady or unsteady have to be made, so as to ascertain whether linearization (of the N.S. equations) to facilitate a solution is valid. As reviewed by Gartshore (5) small steady velocity perturbations permit linearization, and the linearized theory apparently describes most of the field in a rotating fluid (axisymmetric), but practical experimentation can easily illustrate disagreement when the motions increase.

Typical boundary assumptions made in theoretical analyses for steady flow with cylindrical co-ordinates, r, θ, Z; and corresponding velocity components \( V_r \), \( V_\theta \), and \( V_z \) are: \( V_r = 0 \) (radial velocity through cross section reference plane) = 0, tangential velocity \( V_\theta = r \cdot \Omega \) and axial velocity \( V_z = \text{constant} \). Many attempts have been made to try to determine solutions,
considering both small unsteady and large perturbations. Although these may be interesting mathematically they turn out to be of little direct consequence in the practical analysis of motions within and surrounding the free vortex.

Potential flow theory has often proven unrealistic in the analysis of some swirling flow applications, because of the shear stresses present, which for an irrotational spiral vortex become large at small radii, even for values of small viscosity. However, when the region surrounding the singularity (at the point \( r = 0 \)) has been eliminated from the flow field, the theory may still be of some use. Significant work was done by Binnie* and Taylor** in applying the potential theory to an irrotational vortex with an axisymmetric air-core, who maintained that if an allowance is made for the effect of viscosity on the magnitude of swirl, then the flow rate within the vortex will adjust by a definite amount. (c.f. Binnie and Hooking's theory, equation (a) in the discussion below).

Both Binnie and Taylor applied their results to practical configurations. Taylor concluded that the viscous boundary layer effects on the walls of his small atomizer made the potential theory inapplicable, whilst Binnie, in applying the Irrotational Theory to large systems showed that a fair agreement could be obtained. Moreover, this has been well substantiated experimentally by others\(^\text{(9)}\) and is directly relevant to the study of air-entraining vortices in pump sumps.

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Binnie and Hooking's Analysis is outlined briefly as follows:

To derive an expression for calculating the core size and discharge, for a given swirl and depth, the following assumptions were made:

a) the radial velocity at the critical outlet section is negligible, thus implying a vertical core section at this point.

b) the swirl velocity distribution at the outlet is of a free vortex form, and for maximum stability of flow, the outlet quantity is a maximum.

Bernoulli's equation is now applied at two points on the free surface of the vortex (c.f. Fig. 6). Firstly, at a large radius where depth \( H \) is large and the swirl considered negligible, so that

\[
H = K + \frac{v^2}{2g} + \frac{C^2}{2gr^2}
\]

where \( C = V_0 \cdot r \)

and secondly, at the outlet radius \( r \), on the core surface radius \( b \).

The vertical velocity through the outlet is assumed constant. Hence:

\[
K + \frac{v^2}{2g} + \frac{C^2}{2gr^2} = 0 + \frac{v^2}{2g} + \frac{C^2}{2gb^2}
\]

and

\[
k = \frac{C^2}{2g} \left( \frac{1}{b^2} - \frac{1}{r^2} \right)^{\frac{1}{2}}
\]

This substituted back into the first equation gives

\[
H = \frac{v^2}{2g} + \frac{C^2}{2gb^2}
\]

and

\[
V = (2gH - \frac{C^2}{b^2})^{\frac{1}{2}}
\]

but discharge quantity, \( Q = \text{Vertical velocity} \times \text{area} \)
AIR - ENTRAINING SPIRAL FREE VORTEX
AFTER BINNIE & HOOKING'S THEORY.

\[ \text{AIR CORE RADIUS} \quad - \quad b \]
\[ \text{TANK OUTLET RADIUS} \quad - \quad a \]

† A.M. BINNIE & G.A. HOOKING "Laboratory experiments on Whirlpools"
To determine the stable outlet discharge $Q$ must be a maximum.

i.e. \[
\frac{\partial Q}{\partial b} = 0 \quad \text{and} \quad \frac{\partial^2 Q}{\partial b^2} \rightarrow -\text{ve}.
\]

This analysis has been experimentally shown to be applicable to many types of outlet conditions*, with good agreement between the measured tangential free surface velocity and the non viscous distribution. Also, an approximate agreement between the quantity measured and that calculated from the nonviscous theory was reached.

Experimental surface profile comparisons showed the measured outlet area to be in the order of 30% greater than those calculated by relaxation methods**, and hence the need for a coefficient of discharge.

This discrepancy was likely caused by the high viscous shear stresses near to the outlet, which tend to reduce the tangential velocity and thereby increase the radius $r$ as per the free vortex distribution, namely

\[
V_t r = \text{constant}
\]

where $(r)$ experimental $> (r)$ predicted theoretically.

More recent analytical attempts (as described by Gartshore\(^{(6)}\)) have been made by treating the outer and inner edges of the vortex core separately, resulting in a description of the various instabilities, and certain possible analogies to unstable flow between two rotating cylinders.

---

*Quick, "A Study of the Spiral Free Vortex" (Thesis 1961)
Anwar, "Flow in a Free Vortex" (Water Power April 1965)
Other attempts have been made to improve the application of the potential theory, by placing restrictions upon the tangential motions in the radial direction

i.e. let \( V = V_\theta (r) \)

Thus the Navier Stokes and continuity equations now reduce to \( V_z = Z f(r) \) in cylindrical coordinates for steady flow. This general case gives a large variety of possible solutions, which were calculated by Donaldson and Sullivan* for the boundary conditions \( r = R_6 \) and \( z = Z_6 \).

Because of the large variations in the axial velocity distributions (c.f. Fig. 3 and 6) which has several zero points, the flow region was divided into cells, one cell/region between two zero points, and thus a new closed form of solution was achieved for a multi-cellular flow, producing new stream surface profiles. These have not yet been experimentally verified.

There are numerous other theoretical works ranging from numerical solutions of the vortex core and boundary layer approximations, to Momentum Integral methods applied to the linearized theory, and which will not be discussed in this paper.

SECTION IV: PAST AND PRESENT METHODS OF MINIMIZING THE VORTEX PHENOMENA IN PUMP INSTALLATIONS

There are three basic and sequential approaches to the overcoming of the problem of vortex formation:

a) incorporation of design principles eliminating those features which encourage the formation of vortices;
b) preliminary precaution of model testing for large or complex projects;
c) the 'on site' modifications.

In all cases the ultimate aim is to achieve a stable, balanced and uniform flow into the intake's bellmouth.

Subsection IV (a): There have been many proposed lists of rules for good design (10 and Appendix IV), however, it is the understanding of the principles involved which are of primary concern.

To achieve a uniform and balanced flow into the bellmouth, a non-swirl condition must exist within the sump. Thus, good design will try to eliminate any feature that may be a potential source of disturbance such as:

(1) discontinuous or sharply altering surfaces which might lead to separation at the boundary, and cause swirl.

(2) ensure certain minimum clearances between the bellmouth intake and all boundaries, so that the uniformity of the stream line distribution at entry is not disturbed or bunched by the presence of the boundary itself. (c.f. Fig. 7, a & b).
Uniform stream line distribution into bellmouth with adequate submergence and clearance.

Bunching of stream lines accompanying insufficient clearances.

Balanced entry flow

Flow at intake with an unbalanced stream line distribution.

\[(FR)_s = \frac{V}{\sqrt{gS}}\]

Froude submergence number.
Fig. 7 cont.

BALANCED FLOW WITHIN THE BELLMOUTH INTAKE

e)

Ideal streamline distribution $\psi$

Velocity potential lines $\phi$ are perpendicular to the S.L's.
This includes the free water surface boundary, as misjudgment of
the necessary submergence has sometimes led to temporary unsealing of
the bellmouth, particularly in the downsurge starting condition.

(3) the stream line distribution, upon which the minimum boundary
clearances depend, is itself a function of the sump approach
velocity, and the flow rate into the bellmouth intake, and may
sometimes be conveniently represented by the submergence Froude
No. parameter, \( (F.R.)_s = \frac{V_p}{\sqrt{g.s}} \)

where \( V_p \) - velocity of fluid in intake pipe and
\( S \) - the Bellmouth: Free Water Surface distance.

The Bellmouth: Floor Clearance (c) should be such that under
ideal conditions the flow entering the bellmouth would be equipotential
with a perfect stream-line entry distribution. (c.f. Fig. 7 c,d, & e).

(4) the type of sump affects the flow pattern (i.e. terminal or
partial bypass sump).

(5) totally or partially submerged upstream objects tend to induce
alternate eddy shedding, as per the von Karmen vortex phenomena,
which, if the conditions are suitable, may organize themselves into
air-entraining vortices. This is specifically relevant to multi-
pump sump design, where pumps in line with the approach flow may well
cause trouble.

(6) large scale turbulence is known to suppress the formation of vortices,
by destroying their organization, consequently above a certain
channel Reynold's number (and subsequent turbulence level), air-
entraining vortices are not easily formed.
Subsection IV (b): Model testing is discussed in full in Section V.

Subsection IV (c): As a result of necessity, many ingenious devices have in the past been employed for suppressing unwanted vortices in real installations. They are briefly described as follows:

1. The most simple is to raise the free water surface and thereby increase the critical submergence, stretch the vortex core and encourage a better stream-line distribution.

2. The floating of a raft, fixed and placed in such a way as to offer resistance to the surface swirls, and hence dissipate some of the vortex's angular momentum.

3. The floating of a sphere, which gets caught up in the surface depression and drawn towards the vortex core, blocking it and preventing further air-entrainment (rather an extreme remedy).

4. The placing of submerged structures in 'dead' water areas, or areas where vortices are most easily formed. This includes such ideas as turbulence raising baffles; aero-foils built as submerged wall projections; crosses directly beneath the bellmouth inlet; and any other device, that will itself either directly break up the organization of the vortex swirl, or indirectly do so by inducing a high enough turbulence level to dissipate its angular momentum.

5. The bringing of the physical boundaries so close that they no longer permit enough 'organization time' for the vortex to form; and yet do not disturb the stream line distribution unduly.

6. It has been suggested (8) that a deliberately introduced artificial motion centered about the suction intake will inhibit vortex formation in that region. However, such pre-swirl will tend to affect the machines' efficiency.
possibly a more effective and conveniently installed 'remedy' is the expanding metal screen (2), a sophisticated application of Nos. (4), which has of late found practical success. These screens consist of a diamond shaped mesh*, placed end on and parallel to the direction of approach flow. So placed, there is little strain on the screen from the normal approach flow, and any turbulence, swirl or vorticity occurring about the intake will tend to turn into the side of the screen, where the resistance offered is enough to dissipate the rotational movement. Thus, the screen not only inhibits vortex formation, but also acts as a tranquillizer to non-air entraining swirl and turbulence. Furthermore, because the channel mainstream flow is mostly parallel to the screen, the maintenance required to keep the screen from blocking is minimal, and the end-on resistance offered by the screen to the mainstream flow has in practice, been found to be negligible.

SECTION V: MODEL TESTING

General Background

Model testing in any branch of engineering has to be based upon the similarity laws, and hence for complete similarity between model and prototype the following fundamentals must be satisfied:

Geometric Similarity - the direct proportionality between all linear dimensions, model to prototype;

Kinematic Similarity - direct proportionality between velocities; and

Dynamic Similarity - direct proportionality between corresponding forces and corresponding masses.

Basic hydraulic theory shows that the forces acting upon a fluid are:

a. gravitational
b. viscous
c. surface tension
d. elastic forces.

By the simple application of Dimensional Analysis, we can express these forces as dimensionless numbers, though (d) may be neglected, if incompressible fluid flow is assumed. (See Table I).

In practice, sump design is often strongly influenced by the economic limitations and the general structural layout of the building or project which encompasses it. A free water surface will often exist over the greater part of the intake, and consequently gravitational forces must be taken into consideration in this region. (Described by the Froude No.).
### TABLE I
DIMENSIONLESS NUMBERS

<table>
<thead>
<tr>
<th>Type of Force</th>
<th>Dimensionless Nos.</th>
<th>Constant</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gravitational</td>
<td>$\frac{V}{\sqrt{gL}}$</td>
<td>Froude No. (FR)</td>
</tr>
<tr>
<td>Viscous</td>
<td>$\frac{VL}{\mu}$</td>
<td>Reynolds No. (RE)</td>
</tr>
<tr>
<td>Surface Tension</td>
<td>$\frac{V^2L}{\sigma}$</td>
<td>Weber No. (WE)</td>
</tr>
<tr>
<td>Elastic Forces</td>
<td>$\frac{\rho V}{\varepsilon}$</td>
<td>Cauchy No. (NC)</td>
</tr>
<tr>
<td></td>
<td>$\frac{V}{A}$</td>
<td>Mach No. (M)</td>
</tr>
</tbody>
</table>

| Eddy shedding from submerged objects is related to: | $\frac{N_D}{V}$ | Strouhal No. (S) |

where

- $V$ - velocity of fluid
- $a$ - velocity of sound
- $L$ - dimension of length
- $g$ - gravitational constant
- $\rho$ - density
- $\mu$ - viscosity
- $\varepsilon$ - bulk modulus
- $\sigma$ - surface tension
- $N$ - frequency of eddies shed/sec
- $D$ - dia of submerged object
It is generally accepted, that viscous forces (described by the Reynolds No.) assume predominance actually within the suction pipe; whereas in the sump both gravitational and viscous forces are considered to interact. These two similarity requirements give rise to the much discussed classical model paradox of incompatibility.

For example, if \( N \) be the scale ratio between model and prototype. Then to satisfy the Reynolds number similarity criteria, the velocity of flow must be increased by a factor of \( N \). Similarly, for Froudian similarity, the velocity of flow must be reduced by a factor of \( N^{\frac{1}{2}} \), leaving an obvious practical contradiction.

Further, the situation is made more complex by the direct comparison of the vortices themselves. Because of the conservation of momentum in a free vortex, and the dependance of the core transition/formation upon the ratio of the inertia to viscous forces; it is plain to see that if the model is run at a flow rate significantly less than that stipulated by the Reynolds number, the viscosity will have too large an effect. Hence, the high rotational speeds normally experienced at the core of the vortex will not occur. This in turn will result in a lesser pressure reduction within the core, and a reduced surface depression at the vortex. (c.f. Fig. 3).

If, on the other hand, the viscous inertia effects are kept strictly in proportion by increasing the model velocity; the shape of the vortex's surface, dependant upon the ratio of the inertia to gravitational forces, will become exaggerated. Moreover, the determination of the incipient threshold of air-entrainment may not be totally independant of surface tension affects, and so the Weber number could have to be considered.\(^{(2)}\)
Turbulence is also a consideration, since vortex formation is not independent of turbulence intensity. Consequently, surface roughness should be taken into account when selecting materials from which to construct the model, particularly in the case of large scale ratios (i.e. ratios 1:20)

\[
\left(\frac{L}{e}\right)_p = \left(\frac{L}{e}\right)_m \quad \text{where } 'e' \text{ denotes the surface roughness}
\]

It has been observed (4) that entrained air tends to break into bubbles at a definite size, irrespective of scale.*

More recently work has been done (9) on the direct comparison of vortices themselves, using similar models (drain vortex in a cylindrical tank, under idealized conditions) with different scale ratios.

There is good evidence to show that the formation of a stable air-entraining vortex (i.e. one constant with respect to space and time) is dependant upon two distinct mechanisms, namely the generation of vorticity within the boundary layer of fluid approaching the suction pipe, and the shedding of discrete eddies from submerged objects or discontinuous surfaces in the vicinity of the intake.

Though the latter corresponds well with the idealized non-viscous mathematical solutions, and is dependant upon the Froude number, the generation of swirl within the approach flow strong enough to create the vortex, is clearly not independant of the Reynolds number.

*It is likely that bubble formation is related to surface tension, kinematic viscosity and the main stream velocity. By expressing these parameters as a non-dimensional combination of the Reynolds and Weber numbers it is possible to obtain another similarity criteria, namely, a 'Bubble No.', which is of academic interest, and doubtful practical worth:

\[
\text{R.E.} = \frac{VD}{\text{W.E.}} = \frac{\frac{UD}{VD}}{\text{W.E.}} \quad \frac{\text{W.E.}}{\text{V.D.}} = \left(\frac{\sigma}{\gamma V} \right)
\]
Further tests* have shown that by comparing experimental results with the theoretical relationships of a steady vortex flow in a viscous fluid, i.e. boundary layer theory, an apparent increase in viscosity due to turbulent momentum transfer was noted, and agreement between theoretical and practical results could only be reached by incorporating the hypothetical idea of 'apparent eddy viscosity'.

Attempts (2) and (7) have also been made to examine the behaviour of full scale installations of unsatisfactory design. To overcome this has always posed the practical difficulty of obtaining sufficient vortex data as well as the problem of determining the machines actual discharge when it is containing unknown quantities of air, the 'Centre de Chatou' in France (2) achieved reasonable success by rebuilding part of the problematic installation with transparent materials and incorporating photographic techniques.

In the past, thought has been directed to overcoming this model/prototype discrepancy by varying some of the parameters of the fluid itself. (2)

Namely: dynamic viscosity, \( \nu = \mu/\rho \)

specific weight, \( \omega \)

or the surface tension \( \sigma \)

Modification of the Reynolds number by this method could possibly be achieved by either varying the temperature, or by changing the type of liquid used in the model.

*Einstein and Houn Li, "Vortex Motion in Viscous Fluids Studied in Apparatus Consisting of Concentric Glass Cylinders". (Civil Engineering, Sept. 1953).
However, it has been generally concluded that though the requisite variation of one parameter might be achieved by varying the temperature, because all the other fluid parameters are also temperature dependant and do not vary proportionately, the necessary overall vortex similitude could not be achieved.

Similarly, there is as yet no known fluid with which to replace the 'prototype' fluid, within the model, which can offer the other necessary properties.

Though a corresponding modification of the surface tension by the introduction of an auxiliary surface liquid may be achieved practically, it is clear that any auxiliary surface liquid must be immiscible with the main body fluid, since both ultimately have to be recirculated together through the system.

Trends in Model Testing Practice

It is now apparent that though Froudian similarity gives a useful illustration of the flow's stream-line distribution within the regions of vortex formation, it is an insufficient condition on its own to give quantitative results. Also, the fulfillment of the Reynolds number criteria poses practical problems, such as 'shooting' of the approach flow, and extreme turbulence.

Many large industrial organisations, consultancies, etc., have simply abided by the compromise of having the same velocities in both model and prototype. This equal velocity approach is a severe condition and gives a high Froud number, which is often a safe criterion, though offers no guarantee of "perfect similarity" since it may disguise the more subtle vortex mechanisms.
It has been shown (9) that once eddy shedding has commenced, the flow pattern does not alter significantly for any further velocity or Reynolds number increase. Thus, it would seem that providing this eddy shedding condition be met, reasonable flow representation within the model can be achieved. Any difference in turbulence levels likely to suppress the vortex forming mechanisms would have been assimilated by this increased velocity/(R.E.) and which might otherwise not necessarily have been made apparent with the equal velocity procedure.

Optimization of Approach to Model Testing

In the light of the preceding discussion, the most reasonable approach towards model testing would be along the lines suggested by Quick (9), summarized as follows:

a) the model should be operated at a high R.E. number, and as near to that of the prototype as practically possible, whilst the flow pattern is observed.

b) operate the model at successively lower R.E. numbers until the flow pattern is seen to change, the sump main stream velocities become too small to carry on alternate eddy shedding and the flow pattern is seen to change. This fixes the critical R.E. number. (RE_C).

c) if this (RE_C) velocity is less than the FR number velocity, operate the model at the FR number.

d) if the (RE_c) is greater than the FR number, operate the model at the (RE)_c number.

The two major practical difficulties likely to be encountered with this method of experimentation are firstly, the problem of pumping
significantly increased flows from the sump to obtain the high \((RE)_{c}\) value. This may be partially overcome by temporarily increasing the size of the outlet pipe, and possibly increasing the submergence; only to be corrected again when applying the FR test conditions. Should the problem of the sump going into a 'shooting' condition occur, before a constant eddy shedding condition is reached, then it indicates that the model scale ratio is too large to obtain fully representative results. The only satisfactory remedy for this case is to remake the model with a smaller scale ratio.
SECTION VI: DISCUSSION OF PAST AND PRESENT CONCEPTS OF VORTICITY

Summary

In the past, the following have been generally accepted tenets:

a) the forces acting upon a free spiral vortex are gravitational, viscous and surface tension.

b) the conditions for the formation of the hyperbolic part of the vortex core on a free water surface, correspond accurately with Froudian similarity. There is also a good correspondence between practically determined surface profiles and those calculated by non-viscous theory.

c) to obtain similarity between models with different linear scale ratios, the sump main stream velocities should be high enough to simulate corresponding eddy shedding and incipient air-entrainment conditions between the model and the prototype.

d) this in turn, will limit the size of the linear model scale ratio, since turbulence is independent of the scale and the higher velocities within the model could well reach a turbulence level capable of vortex suppression, before this eddy shedding condition had been reached, where as this would not necessarily occur in the prototype.

e) in the past, vorticity experienced in pump sumps has been loosely categorized as occurring in three forms, namely local, drain and column vortices.

f) swirl has long been considered a cause of vorticity and has even been classified into three distinctive phases.

The degree of swirl was seen to be a function of both the main stream velocity, (expressed as the product of the Reynolds number
and the boundary shear stress represented by the D'arcy coefficient of friction in the form $R = \frac{N_d}{\alpha}$, the suction pipe velocity $V_a$ and diameter $d_p$, (expressed by the parameter $\tan \alpha = \frac{N_d}{V_a}$)

g) a correspondence has been drawn, between the types of vorticity and the three phases of swirl. (c.f. Fig. 5).

Discussion

Past Concepts:

1. It has been established quantitatively that the measured free surface hyperbolic profile of the vortex core is in good agreement with the cylindrical theory, except at the outlet portion (c.f. (1) and (9)).

2. Vorticity is a function of the Reynolds number, both in the approach channel and within the actual vortex itself. However, whether the affect (represented by the Reynolds number) is significantly instrumental in the formation of the vortex, through the mechanism of eddy shedding and/or boundary layer diffusion formation, or by its presence within the core, is not clearly established.

3. It is plain to see that the three categorizations of vortices as proposed by Young, (10) are effectively one and the same thing. The 'local' vortex is a free spiral vortex starting at the water surface and then drawn through $180^\circ$ up into the mouth of the intake; whereas the "drain vortex" is the same only flowing downwards under gravity.
The 'column vortex' is just a more severe case of the 'local vortex' occurring concentrically about the intake and involving a larger proportion of the sump's flow.

These classifications tend to make the phenomena appear more complicated than they need be.

4. The division of swirl into its three phases by Hattersley, is interesting, though the term 'swirl' itself is misleading and is not easily defined qualitatively. Hattersley's parameter of 'induced swirl', namely \( \tan \alpha - \frac{N_d}{V_p} \), represents the combined axial and tangential velocities of the air-entrained vortex: the average suction pipe's axial velocity.

However, it is unlikely that this parameter could have been measured very accurately, since a freely pivoted meridional vortometer was used, and for the sensitivity required for small scale readings (as in model tests), induced auto-rotation of the vanes could affect the revolution count. Further, if air entrainment did occur, the actual flow entering the intake \( (V_p) \) would be difficult to determine.

5. After performing a series of tests, Denny and Bonnington* comment that it was seen that swirl seldom exceeded about 5° outside the region where air-entraining vortices were operative; whereas, severe swirl leading to a concentric vortex 30°.

*Some measurements of swirl in pump suction pipes (RR 556, Feb. 1950).
From this one must reasonably deduce that the greater the angle of swirl, the greater the severity of the vortex and the higher the chance of air-entrainment will have of occurring.

Hattersley's results do not concur with this, for though he deduces that air-entraining vortices are a function of both \( \tan \alpha \) and the Reynolds number, his experimentations indicated that the most severe swirl, giving rise to concentric vorticity was found to occur at B (Fig. 5), where a high \( \text{Re}/f \) and a low \( \tan \alpha \) exist, rather than the more likely condition of a high \( \text{Re}/f \) and high \( \tan \alpha \).

These results may well have occurred because of the inherent inaccuracies of the vortometry; namely that a free vortex with its centre displaced from the centre of the vortometer inlet tube can pass through the blades of the vortometer without inducing any rotation. Also, because the vortometer is subjected to a high axial velocity component, the flat blades can be subjected to lift forces which produce auto-rotation.

Denny also noted that by drawing some of the main stream flow out of the end of the sump, the vortex formations were significantly reduced. This concurs with Hattersley's findings of a high \( \text{Re}/f \) though seems contrary to the fact that the amount of vorticity generated within the boundary layer is a direct function of the mainstream velocity and hence an increase in the velocity (as above) would increase the vorticity, not reduce it and supposedly it's effect.

A possible explanation is that an increase in the sump's mainstream velocity reduces the organization time and tends to sweep the vorticity past the inlet.
This explanation will be further clarified in the following section after work done by Quick. \(^{(15)}\)

**Present Concepts** \(^{(15)}\) \(^{(16)}\)

All vorticity is generated within the boundary layer and is associated with a boundary shear, the only mechanism by which vorticity is formed.

For practical convenience, it is logical to subdivide this mechanism into two parts: that vorticity generated within the continuous boundary layer flow, and that vorticity shed as discrete vortices from boundary layer discontinuities.

As pointed out by Quick \(^{(15)}\), this division is significant in terms of the organization of vorticity.

When a vortex is shed from an abruptly discontinuous boundary, it is already highly organized, and if the intake be close and the intake velocity high enough, air-entrainment may occur.

Moreover, vorticity generated within the continuous boundary layer may be similarly troublesome and unstable. If the sump's geometry is symmetrical, then a bi-stable situation often arises, with vortices occurring at infrequent intervals.

The actual formational mechanism is best described by Kaufmann \(^{(16)}\): "Consider the Boundary Layer/Pressure distribution diagram"(Fig. 8a). Fluid particles moving in the boundary layer are decelerated because of friction, so that their kinetic energy is reduced.

*Separation is considered here to be boundary shear.*
a) **BOUNDARY LAYER PROFILES ILLUSTRATING THE EFFECTS OF POSITIVE AND NEGATIVE PRESSURE GRADIENTS.** (After Kaufmann)

\[
\frac{\partial u}{\partial y} > 0 \quad \frac{\partial^2 u}{\partial y^2} > 0 \quad \frac{\partial^2 u}{\partial y^2} = 0 \quad \frac{\partial^2 u}{\partial y^2} < 0
\]

\[
\frac{\partial^2 u}{\partial y^2} < 0 \quad \frac{\partial^2 u}{\partial y^2} = 0 \quad \frac{\partial^2 u}{\partial y^2} > 0 \quad \frac{\partial^2 u}{\partial y^2} > 0
\]

b) **VORTEX SLIP-PLANE.** (After Kaufmann)

c) **ORGANIZED VORTEX FORMATION BEHIND A DISCONTINUOUS BOUNDARY.**
If there is a pressure drop in the direction of flow \((\frac{dp}{dx} < 0)\), then the pressure can overcome the deceleration of the particles and they will continue to move along the wall. A positive pressure gradient, on the other hand, will tend to decelerate the particles in the Boundary Layer even more. As a result, the thickness of the boundary layer tends to increase and the particles in it slow down, stop and even possibly reverse their direction of motion (Fig. 8a, Point b).

A reverse-flow layer will tend to push itself between the body surface and the Boundary layer and in this way 'separates' it from the body. A slip plane (c.f. Fig. 8) is created between the two layers and if unsteady, degenerates into vortices which are carried away by the flow. The energy required to produce these vortices cannot be recovered and causes additional resistance between the body and fluid.

It is well known from vortex theory, that vortices propel each other because of the presence of induced flow fields. The stability of the vortex depends upon the direction of the propulsion, which in turn depends upon which boundary surfaces have the most significance with respect to the main stream flow.

If the vortices are propelled in a direction downstream with the flow, they will travel faster, i.e. go further, or need more time, than those propelled upstream against the flow, before they become organized entities.

If the velocity of propulsion upstream against the flow is strong enough to equal, or nearly equal the mainstream velocity, then the vortex may remain in a constant or nearly constant position with respect to the sump. In this case, if it does so for a period long.
enough for it to gather sufficient angular momentum, air-entrainment may occur, which, once started has a tendency to persist. (See later).

Hence, it now becomes apparent why both Denny\(^4\) and Hattersley \(^7\) found an apparent reduction in the vorticity formed within the sump when the main stream velocity was increased.

The amount of vorticity generated increased with the increase in velocity, but was swept downstream past the intakes, before it had time to become organized. As already mentioned, the \(\tan(\alpha_c)\) parameter was suspect, due to probable induced autorotational errors, and because the vortex moves with high vorticity can pass between the blades without causing rotation.\(^{15}\)

### The Persistence of a Vortex and Its Relationship to Energy Levels and Surface Tension

It is of interest to reconsider the core comparisons discussed in Section I and to compare the hypothetical energy levels of a partially developed air entraining spiral vortex.

Let the energy at the surface of the 'free vortex' at the point 'A' be \(E_A\). By applying Bernoulli's equation, we may write:

\[
E_A = \frac{V_A^2}{2g} + \frac{p_a}{f} + h_a
\]

where \(V_A\), for a free spiral vortex, is a function of the fluid's tangential and axial velocity at point (A), such that: \(V_A^2 = V_t^2 + V_a^2\)

Similarly, let the energy at a coincident point now called "B", but within a closed viscous core, be \(E_B\), which is defined as:
\[ E_B = \frac{v_B^2}{2g} + \frac{p_b}{\rho} + h_b + (\text{viscous dissipation losses } C_1). \]

where \( r_a = r_b \)

By inspection (c.f. Fig. 9) we can see that:
1) \( P_a = \text{atmospheric pressure} \)

and ii) \( P_b \approx \text{atmospheric pressure} - Pa \)

If we assume the strength of the two vortices at a given instant to be constant, and similarly the energy, such that: \(- E_A = E_B \)

then:
\[ \frac{v_A^2}{2g} + \frac{p_a}{\rho} + h_a = \frac{v_B^2}{2g} + \frac{p_b}{\rho} + h_b \quad (L) \]

where the viscous shear losses \((\ )\) are large and a function of the viscosity and the velocity gradient across the vortex. (c.f. Fig. 3).

1.e. \[ \frac{v_A^2}{2g} \gg \frac{v_B^2}{2g} + (\mu \frac{dv}{dx}) \]

clearly, in this case \( V_a \gg V_b \). Thus, to effect the transition from a forced fluid vortex core to that of an air core, both within the same free vortex field, would clearly require a considerable increase in severity of the vortex, such that \( E_B \rightarrow (E_B^1) \)

and \[ \frac{v_b^2}{2g} + (\mu \frac{dv}{dx}) \rightarrow (V_b^1)^2 \quad + \quad (\mu \frac{dv^1}{dx}) \]

where \( \frac{dv^1}{dx} \gg \frac{dv}{dx} \) (throughout the field)

Hence \( E_B \) (forced) \( \gg \) \( E_A \) (free) for
\[ V_B \) (forced) \( \gg V_A \) (free) \]

(c.f. Fig. 9)
SUPERIMPOSITION OF TWO VORTICES.

\[ h = \text{atmospheric pressure} \]

\[ h_b = h_a + h_b' \]

\[ r = r_a = r_b \]

A - Surface point of 'A' on spiral free vortex.

B - Point within the fluid body of a forced vortex field.
Because of this energy gradient, it is easy to see that once an air-entraining core has formed, it will persist until the energy level within the region has fallen back below that necessary to maintain the free vortex velocity distribution, with the pressure at the given radius (point B) exactly equal to atmospheric pressure. Thus, with a decrease in severity of the same air-entraining vortex, the energy level at the coincident point will have to fall by an amount equal to \((E_B - E_A)\) below its original transitional level \(E_B^1\) before the air core will collapse, resulting in the vortex's apparent tendency to persist.

Though the effect of the surface tension forces acting along the vortex's free surface remains specifically unknown, (presumably small), it would, in the light of the above, now seem unlikely to be a significant feature with respect to an explanation of persistence; since the very mechanism of surface tension tends to oppose any further surface deformation; whether it be from a plane free surface to an air-core funnel, or vice-versa. This suggests that an equal amount of additional energy would be required to overcome its effect, in either the cases of formation or collapse.

However, it is possible that the surface tension may well affect the initial air-entrainment starting point, as would be expected when using two different fluids under identical pumping conditions.
CHAPTER II

"AN EXPERIMENTAL STUDY OF THE CONTROL OF VORTEX FORMATIONS WITHIN A MULTIPLE IN-LINE PUMP SUMP CONFIGURATION"

SECTION I: OBJECTIVES & DESIGN OF THE EXPERIMENTATION

In the past, many types of layouts have been used for multiple pump sumps, and are illustrated in such papers as (2) (3) (4) (7) (8) (10) and (12).

We will consider the smallest possible sump configuration which is often economically the most desirable, especially in underground installations. The 'In-Line' configuration, such as in Fig. (13), with the flow approaching from one limited direction, possibly from some screening device, might offer the best solution. Experience has shown however, that such an arrangement can give rise to quite severe vortex formations.

The purpose of the present tests has been restricted to studying this 'in-line' layout in detail, in the hope that an understanding of the various vortex phenomena may be used to turn this poor configuration into a satisfactory one, attractive from the economic viewpoint.

Different types of flow guides or shrouds were introduced around the bellmouth intakes, to try to prevent the formation of the Von Karmen Vortex Streets by physically suppressing the eddy shedding from the adjacent upstream pumps.
Also, the sump was tapered by varying the angle between the channel walls to reduce the deceleration within the sump in the direction of flow, and further decrease the formation of vortices.*

Hot film anemometry techniques to monitor the turbulence levels were used to compare the condition of the flow entering the bellmouth intakes.

*cf. Fig. 8, Chapter I, Section IV.
SECTION II: THEORY: RELEVANT TO THE EXPERIMENTATION

The general form of the Navier Stokes equations of motion of a viscous incompressible fluid:

\[ \rho \left( \frac{\partial u_i}{\partial t} + u_\alpha \frac{\partial u_i}{\partial x_\alpha} \right) = -\frac{\partial P}{\partial x_i} + \rho \frac{\partial \Omega}{\partial x_i} + \mu \nabla^2 u_i \]

(where \( u \) and \( \Omega \) refer to co-ordinate axes of reference): may be modified by representing the viscous forces in terms of pressure from elemental consideration within the fluid flow such that

\[ \rho \left( \frac{\partial u_i}{\partial t} + u_\alpha \frac{\partial u_i}{\partial x_\alpha} \right) = -\frac{\partial P}{\partial x_i} + \rho \frac{\partial \Omega}{\partial x_i} + \left( \frac{\partial P}{\partial x} \right)_\alpha \]

By adding the continuity equation to the RHs, where the continuity equation \( \frac{\partial u_i}{\partial x} = 0 = -\rho u_i \left( \frac{\partial u_i}{\partial x} \right)_\alpha \)

we can write the Navier Stokes equations as:

\[ \rho \frac{\partial u_i}{\partial t} + \rho u_\alpha \left( \frac{\partial u_i}{\partial x_\alpha} \right) = -\frac{\partial P}{\partial x_i} + \rho \frac{\partial \Omega}{\partial x_i} + \left( \frac{\partial P}{\partial x} \right)_\alpha - \rho u_i \left( \frac{\partial u_i}{\partial x} \right)_\alpha \]

or \( \rho \frac{\partial u_i}{\partial t} = -\frac{\partial P}{\partial x_i} + \rho \frac{\partial \Omega}{\partial x_i} + \frac{\partial P}{\partial x_\alpha} + \rho \left( -u_i \left( \frac{\partial u_i}{\partial x} \right) - u_\alpha \left( \frac{\partial u_i}{\partial x_\alpha} \right) \right) \)

\[ = -\frac{\partial P}{\partial x_i} + \rho \frac{\partial \Omega}{\partial x_i} + \frac{\partial P}{\partial x_\alpha} - \frac{\partial}{\partial x_\alpha} \left( \rho u_i u_\alpha \right) \]

\[ = -\frac{\partial P}{\partial x_i} + \rho \frac{\partial \Omega}{\partial x_i} + \frac{\partial}{\partial x_\alpha} \left( P_\alpha - \rho u_i u_\alpha \right) \]
Now, if the mean flow is steady, we can write:

\[ u_i = \bar{u}_i + u'_i \]

which is the same as

\[ u_\alpha = \bar{u}_\alpha + u'_\alpha \]

Where \( \bar{u} \) = mean value of the flow velocity

\( u' \) = fluctuating component about the mean

\( u = the \ instantaneous \ velocity \ value. \) (c.f. Fig. 10).

Hence

\[ \rho \bar{u}_\alpha u_i = \rho (u_\alpha + u'_\alpha) (\bar{u}_i + u'_i) \]

and

\[ \rho \bar{u}_\alpha u_i = \rho (u_\alpha + u'_\alpha) (\bar{u}_i + u'_i) \]

\[ = \frac{(u_\alpha \bar{u}_i + u'_\alpha \bar{u}_i + u'_\alpha \bar{u}_i + u'_\alpha u'_i)}{(u_\alpha \bar{u}_i + u'_\alpha \bar{u}_i + u'_\alpha \bar{u}_i + u'_\alpha u'_i)} \]

but the mean of a variation about a mean \( \bar{u}' \) = \( \bar{u}' \) = 0

Similarly

\[ \bar{u}_\alpha u'_i = \bar{u}' \bar{u}' \alpha = 0 \]

Hence

\[ \rho \bar{u}_\alpha u_i = \rho \bar{u}_\alpha \bar{u}_i \]

or

\[ \rho \bar{u}_\alpha u_i = \rho \bar{u}_\alpha \bar{u}_i \]

Hence the Navier Stokes equations now may be written:

\[ -\rho \frac{\partial \bar{u}_i}{\partial t} = -\frac{\partial P}{\partial x_i} + \rho \frac{\partial \bar{\rho}}{\partial x_i} + \frac{\partial}{\partial x_\alpha} \left( \bar{P}_\alpha - \rho \bar{u}_i \bar{u}_\alpha \right) \]

where \( -\rho \bar{u}_i \bar{u}_\alpha \) are often known as the REYNOLDS STRESSES, caused by turbulence.

If the turbulence may be considered homogeneous (c.f. Appendix VI), then the fluctuating components \( u_i \) and \( u_\alpha \) will be equal for all values of \( i \) and \( \alpha \). Thus, the Reynolds Stresses become equivalent to

\[ \rho \bar{u}_i^2 = \rho \bar{u}_\alpha^2 \]

and the deviation of the fluctuating components from the mean main flow can be taken as a measure of the magnitude of the turbulence, and be represented by the dimensionless turbulence intensity parameter (T.I.).
Where T.I. = \frac{\sqrt{(u')^2}}{u}

This parameter may be determined experimentally by the application of hot film anemometry, which can measure independently the $|\langle u' \rangle|$ and $\sqrt{\langle u'^2 \rangle}$.
**Turbulent Velocity Components**

At any point:

\[ u = u' + \bar{u} \]

- \( \bar{u} \) - mean velocity.
- \( u' \) - fluctuating velocity component.
- \( u \) - instantaneous velocity value.

---

**Cylindrical Coordinate System**

- \( Z \) axis
- \( X \) axis
- \( Y \) axis
- \( r \)
- \( \theta \)
- \( V_r \)
- \( V_{\theta} \)
- \( V_z \)
SECTION III: EXPERIMENTAL PROCEDURE

The physical sump-pump dimensions were laid out as close to the recommended standard practice (13) and (14) as possible.

These were:

- Sidewall: Bellmouth - D/2
- Backwall: Bellmouth - D/2
- Sump Floor: Bellmouth - D/2
- Bellmouth Submergence - D
- Distance between pumps - 3D

where D is the bellmouth diameter (c.f. Fig. 11), maximum model flow velocity - 20″/sec (1.75 ft/sec). Normal prototype velocity range - (2-3 ft/sec).

The average mainstream velocity (U) within the sump and its fluctuations (dU) were read from the linearized anemometer module voltmeter.

An external R.M.S. meter connected to the linearized output, enabled the turbulent velocity perturbations (u) to be monitored. Similarly, with the aid of a frequency suppression module, a simple spectral frequency analysis was performed.

A series of probe traverses at two different depths were taken across the sump. The probe sensor was held parallel to the \( \frac{L}{2} \) of the sump and in line with the direction of the main stream channel flow.

To compare the stabilizing effects of differently accelerating flows within the sump, four different sump angles \( \alpha \) were tried:
SHROUDDLNG DESIGNS.

Suction tube diameter (O.D.) = $2\frac{3}{8}'' = d$

Bellmouth diameter (O.D.) = $4\frac{3}{4}'' = 2d$

a) Full Shroud  $R = d$, $z = 3d$  (No bottom).

b) Partial Shroud  $R = \frac{d}{2}$, $z = 2d$  (No bottom).

c) Full Shroud and Bottom  $R = d$, $z = 3d$  (Bottom).

d) Mesh Screen – same dimensions as full shroud, but constructed with fine Cu. Diagonal mesh inplace of galvanized sheet steel.  (No bottom ).
To compare the effects of physical suppression on the formation of vortices shed from the submerged upstream intakes, four types of rigid shroudings were fixed to the intakes and tested at the different sump angles. (c.f. Fig. 11).

The readings taken were tabulated (as per the result section) and the results compared graphically.

**Apparatus - Instrumentation**

A large painted rectangular flume with flow stabilizing screens at each end was used.

Three independently controlled centrifugal pumps drew water from a test channel within the flume and recirculated it back into the approach end of the flume. (c.f. Fig. 12).

The test channel was of a rectangular cross-section, with one moveable wall and channel-end board permitting the sump angle (\( \alpha \)) to be varied. (c.f. Fig. 13). Each pump drew independently through a standard bellmouth piece, vertical intake pipe and semi-flexible suction piping system. Each intake was mounted on its own moveable trolley suspended across the sump, permitting lateral, longitudinal and vertical freedom of movement within the sump. (c.f. Photo 1).

Orifice plates were installed in each suction line enabling flow measurements to be monitored using Hg. manometers.

**Anemometry**

A T.S.I. hot film anemometer with a temperature compensating probe was used to measure the various velocity characteristics within the sump.
APPROXIMATE SUMP LAYOUT

SECTION ALONG X-X:

- Sump angle
- F.I. - Flexible intake pipe
- S - Flow screens
- W.S. - Movable wall supports
- M.W. - Movable wall
- B - Bellmouths
- P - Pump
- M - Motor
- V - Discharge valves

PLAN:
Fig. 13

TEST CHANNEL SECTION.

Fixed channel wall

S = 5"
D = 4\frac{3}{4}"
d_p = 2\frac{3}{8}"

3^{rd} bar position-probe traverse line
2^{nd} bar position-probe traverse line
1^{st} bar position-probe traverse line

Sump angles
\alpha = 12^\circ, 20^\circ \text{ and } 24^\circ

Flow
Movable channel wall
The anemometer probe/sensor was held in an adjustable, sliding, mounting block, permitting a traverse of readings across the sump. (c.f. Photo I)

**Principles of the Hot-Film Anemometer**

A pre-calibrated, internally heated sensor is kept at a constant temperature and greater than that of the ambient fluid to which it is exposed. Any relative motion between the fluid and sensor results in a change in the heat transferate between the two, with a tendency to cool the sensor. The anemometer automatically compensates and monitors the additional electrical energy necessary to maintain the probes constant temperature, and this is read off on the anemometer module voltmeter. The signal was linearized and easily converted from volts into ins/sec by a simple calibration curve. (c.f. Graph No. I).

An external R.M.S. meter connected to the linearized anemometer output enabled the R.M.S. of the turbulent velocity perturbations of the flow to be monitored. Also the same anemometer signal fed into a suppression module with built in high and low pass variable frequency filters, enabled a simple spectral frequency analysis to be performed. (c.f. Fig. 14).
A) T.S.I. PROBE CIRCUITRY.

B) PROBE & HOLDER IN TEST CHANNEL

PHOTO 1
APPARATUS & INSTRUMENTATION SPECIFICATION

Flume Dimensions: 25' x 7' x 3 3/4'.

Pumps: 3 x 3/4", 1,800/rpm, Capacity — 578 ins/sec.

Motors: 3Ø, 600/s, 1,800/rpm, Class B insulation, 230v/s, total 3.5 amps.

D by D/2 Orifice Plates: D — 7 1/2" angle — 38°
    d — 2 1/2" thickness — 1/4"

Orifice plate coefficient

Bellmouths: Smallest I.D. — 2 3/8" (plexiglass)
            Largest I.D. — 4 3/4"
            Depth — 2 3/8"

T.S.I. Anemometry: Anemometer module model No. — 1050
                   Linearizer module model No. — 1055
                   Signal Suppress module model No. — 1057
                   Correlator module model No. — 1015c.

(T.S.I.) Probe — wedge type, temperature compensation,
     model No. — 1336E.
     Range 0-3m/sec, overheat ratio: 1.07,
     Position of coarse span control — 8.
     Position of fine span control — 7.

Probe calibrated for Bridge output at zero flow — 4.25v/s.
Probe calibrated for Bridge output at full flow — 13.60v/s.
SECTION IV: RESULTS

Readings Recorded

\( \bar{u}' \) - the mean of the external R.M.S. meters' fluctuating values \( (u'_a) \) and \( (u'_b) \), representing the high frequency turbulence perturbations

where:

\[
\bar{u}' = \left( u'_a + u'_b \right)^{\frac{1}{2}}
\]

\( \bar{U} \) - the anemometer module voltmeter's mean of the fluctuating values \( (U_A) \) and \( (U_B) \), representing the average main stream flow velocity

where:

\[
\frac{U_A + U_B}{2} = \bar{U}
\]

\( (dU) \) - computed from the main stream flow fluctuating velocity values, such that \( dU = (U_a - U_b) \)

T.I. - the turbulence intensity values, computed for each probe position,

where \( T.I. = \frac{\bar{u}'}{\bar{U}} \)

\( L \) - Probe traverse distance from the fixed channel wall. (Fig. 13).

For each traverse position \( (L) \), readings were taken at two different depths, with the sensor submerged in a line:

(a) approximately half way between the channel floor and bellmouth ring \( (s \sim 6") \), so as to indicate the stream-line states as they approached the intake.

(b) approximately half way between bellmouth ring and the free water surface \( (s \sim 4") \), to indicate the state of the top half flow, and the likelyhood of air-entraining vortices forming.
Submergence Froude Number

\[ (FR)_s = \frac{V_p}{\sqrt{gS}} \]

where \( V_p \) - intake pipe velocity (ins/sec)
\( S \) - bellmouth submergence depth (ins)
\( g \) - 32.2x12 (ins/sec)

\[ (FR)_s = \frac{114.6}{32.2 \times 12 (s)} = 2.6 \]

Submergence Froude Number
### T.S.I. Anemometer Calibration Curve

**Graph No. 1**

**PROBE No. 2**

<table>
<thead>
<tr>
<th>Flow</th>
<th>Set volts</th>
<th>Linearized output deflection</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>4.25</td>
<td>0</td>
</tr>
<tr>
<td>full (3m/s.)</td>
<td>13.60</td>
<td>300</td>
</tr>
</tbody>
</table>

1 m/s. = 39.4 ins./s.

1 1/2 m/s. = 59.1 ins./s.
ANEMOMETER PROBE, STABILITY TESTS

First calibration test showing effects of dirt & bubble accumulation on probe. Detergent 'ALCONOX' added to test water.

Sump cleaned + fresh water at 60°F. Experimentation with methyl hydrate solvent added to water and probe vibrated.

Final procedure: before each reading, probe cleaned with solvent and vibrated to free bubbles. Clean water at 70°F.

R. M. S. $\bar{u}$

Time scale = 0.01 secs.

$\bar{u}$' accurate to ± 0.004 vs.
FIRST BAR POSITION.

3D upstream of first pump.
SECOND BAR POSITION

Sensor's Submergence = 4".

- --- NO SHROUDING
- --- FULL SHROUD + BOTTOM
- --- MESH SCREEN

Traverse (ins.)
SECOND BAR POSITION.

Graph No. 5

Sensor's Submergence = 4'.

Traverse (ins.)
SECOND BAR POSITION.

Sensor's submergence = 6".

Graph No. 6
SECOND BAR POSITION.

Graph No. 7

Sensor's submergence = 6".

- - - - NO SHROUDING
- + + + FULL SHROUD
- △-△- PARTIAL SHROUD

Traverse (ins.)
THIRD BAR POSITION.

Sensor's submergence = 4".

Graph No. 8

Traverse (ins.)

NO SHROUDING

MESH SCREEN

FULL SHROUD + BOTTOM
THIRD BAR POSITION.

Graph No. 9

Sensor's submergence = 4".

α\(\sim\) 24° --- △---△--- PARTIAL SHROUD
α\(\sim\) 24° --- +--- +--- FULL SHROUD
α\(\sim\) 24° --- --- --- NO SHROUDING

Traverse (ins.)
THIRD BAR POSITION

Sensor's submergence = 6"

\[ \alpha \sim 20^\circ \quad \circ \quad \text{FULL SHROUD + BOTTOM} \]

\[ \alpha \sim 24^\circ \quad \square \quad \text{MESH SCREEN} \]

\[ \alpha \sim 2^\circ \quad \text{NO SHROUDING} \]
Graph No. II

THIRD BAR POSITION.

Sensor's submergence = 6"

\[ \alpha \approx 24^\circ \]  
- Triangle with diagonal line: PARTIAL SHROUD
- Plus sign: FULL SHROUD
- Dash line: NO SHROUDING

Traverse (ins.)

[Graph depicting sensor's submergence and traverse positions with various lines for different submergence conditions and shrouding scenarios.]

[Diagram showing data points and trend lines for sensor behavior under different conditions.]
TYPICAL PLOT OF TURBULENCE INTENSITY ACROSS SUMP

AT $\alpha = 24^\circ$.

Second Bar position
8" Traverse submergence

- $\Delta \Delta \Delta$ PARTIAL SHROUD
- $+++$ FULL SHROUD
- $+++$ NO SHROUDING

Graph No. 12
TYPICAL PLOT OF TURBULENCE INTENSITY ACROSS SUMP AT $\alpha = 20^\circ$.  

Third Bar position  
6" Traverse submergence  

---  
- MESH SCREEN  
- FULL SHROUD + BOTTOM  
- NO SHROUDING  

Turbulence Intensity ($\overline{U}'$)  

Sump Traverse
The accuracy of the experimental readings taken was estimated to be:

\[(\bar{U}) - \text{to within 2 volts (lins/sec)}\]
\[(u') - \text{to within 0.004 vs (lins/sec)}\]

It was seen, that as the channel flow approached the bellmouth intakes, the values of \((u')\) increased with increase in the central streamline distortion, whilst \((dU)\) increased in the same central region due to the eddy-shedding from the upstream pumps and the values of \((\bar{U})\) declined unexpectedly. The measured value of \((\bar{U})\) occur because either the magnitude of \((\bar{U})\) actually decreased as the flow approached the intake, or because of an inaccurate response of the sensor in this region of prevailing flow.

Observations indicated that both possibilities occurred, as illustrated by Figs. 15, 16 and 17; for unlike the hot wire anemometer, the unidirectional hot film sensor is still sensitive to flows impinging upon it at angles other than in it's central line of direction. Thus, it is possible for two flows of the same velocity impinging upon the same sensor at different times, to result in different monitored values of \((\bar{U})\). (Fig. 16 a, b, & c).

Similarly, two vortices of the same strength and sense of rotation would register different values of \((\bar{U})\) in a way proportional to their central distances from the sensor. (Fig. 17 a, b, & c).

It is now clear that, even by assuming HOMOGENEOUS turbulence*, a unidirectional sensor, in a three-dimensional unsteady flow, will not accurately monitor the velocity \((\bar{U})\) nor ratio \(\frac{u'}{U}\). Hence for purposes of these tests, the turbulence parameter was \(\frac{u'}{U}\) abandoned.

*(c.f. Appendix No. V)
STREAM LINE DISTRIBUTION ABOUT
2-IN-LINE PUMP BELLMOUTHS.

Region of reduced stream line velocity

Line of probe traverse

Main stream flow

\[ \alpha/2 \]
Fig. 16

EFFECT OF DIFFERENT FLOW INCIDENCE ANGLES ON A UNI-DIRECTIONAL HOT FILM ANEMOMETER SENSOR.

a) \( \bar{U}_0 \pm u_0' \)

b) \( \bar{U}_1 \pm u_1' \)

c) \( \bar{U}_2 \pm u_2' \)

where: \( \bar{U}_0 \pm \bar{U}_1 \pm \bar{U}_2 \) and \( u_0' = u_1' = u_2' \) - Isotropic turbulence

d) \( \bar{U} = \bar{U}_x + \bar{U}_y \)

\( \pm v = \pm v_x = \pm v_y \) - Isotropic turbulence

N.B. The direct effects of \( \bar{U}_y \) on sensor are unknown.


\( \bar{U} \) - Average velocity of the mean mainstream flow (\( U \)).

\( u_1' \) - R.M.S. of the turbulent fluctuating velocity components (\( \pm v \)).
EFFECT OF ORGANIZED VORTICITY UPON A UNI-DIRECTIONAL SENSOR’S RESPONSE.

A free vortex distribution \((V_T, r = \text{constant})\) is assumed, such that \(\langle V_T \rangle_B = \langle V_T \rangle_C\), where \(U_B < U_C\) and the distance \(B > \text{distance} \ C\). The effect of the velocity component perpendicular to the sensor’s \(\theta\) is significant but unknown.

\[
A_{A1} = U + P - R^2 \quad A_{A2} = U + P + R^2
\]

But monitored value from anemometer = \(|R_A| > |R'|\)

\[
A_{B1} = U + P + R' \quad A_{B2} = U - P + R'
\]

When \(|y| > |y'|\), \(A_{C1} = U + P + (x) + (y')\) and \(A_{C2} = U - P - (x) + (y')\)

\(R'\) = Anemometer recording of \(R\) where \(|R| > |R'|\).
\(B\) and \(C\) = \(\theta\) Displacements between sensor and vortex.
\(A\) = Resultant velocity along the probe sensors \(\theta\).
\(P\) = Propulsion velocity of vortex.
\(R\) = Abs. velocity at a point in the fluid.
\(U\) = Average main stream velocity.
\(A\) = Anemometer.
A simple visual spectral analysis indicated two distinct 'Scales of Turbulence' to be present, with differing amplitudes and frequencies. The energy losses due to the small amplitude and higher frequency velocity turbulent fluctuations (10-200 c/s) were represented by \( \overline{u}' \) and considered to be significantly smaller with respect to the pumps than those caused by the lower frequency, larger amplitude fluctuations brought about by organized vortices passing the sensor and represented by \( \overline{dU} \). Thus, though \( \overline{dU} \) may not be considered as an absolute measurement, it may be taken together with \( \overline{u}' \), to reasonably represent the measure of badness (if large) or goodness (if small) of the state of flow in the sump.

Because of the large number of readings taken across the sump, and because the physical layout of the sump was kept unaltered, each reading could be assumed to have a similar 'Error Probability' and therefore, be accurate enough for the comparative purposes used in these tests.

**Significance of the Results**

Table 2 shows the comparison of the results in order of preference, where the worst reading (i.e. largest value of \( \overline{dU} \)) was recorded for each depth and traverse, and is listed for each type of shrouding tested. All types of shrouds tested, were shown to be an improvement, when compared to the same sump situation without the shroudings. As expected*, an increase in the sump angle \( \overline{\alpha} \) bringing about a positive channel acceleration, improved the flow conditions, by reducing the

*From discussion Chapter I, Section IV.
### TABLE 2

**COMPARISON OF RESULTS**

<table>
<thead>
<tr>
<th>Order of Preference</th>
<th>Type of Shrouding</th>
<th>2nd Bar Traverse $dU$ (volts)</th>
<th>3rd Bar Traverse $dU$ (volts)</th>
<th>Sump Angle ($\angle$)</th>
<th>Streamline Interference</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>4&quot; subm. 6&quot; subm.</td>
<td>4&quot; subm. 6&quot; subm.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>Partial Shroud</td>
<td>10 7</td>
<td>6 7</td>
<td>24°</td>
<td>Moderate</td>
</tr>
<tr>
<td>2</td>
<td>Mesh Screen</td>
<td>10 8</td>
<td>8 7</td>
<td>20°</td>
<td>Minimum</td>
</tr>
<tr>
<td>3</td>
<td>Full Shroud</td>
<td>$8\frac{1}{2}$ 10</td>
<td>11 $6\frac{1}{2}$</td>
<td>24°</td>
<td>Large</td>
</tr>
<tr>
<td>4</td>
<td>Full Shroud &amp; Bottom</td>
<td>11 12</td>
<td>4 4</td>
<td>20°</td>
<td>Maximum</td>
</tr>
<tr>
<td>5</td>
<td>No Shroud</td>
<td>16 15</td>
<td>16 13</td>
<td>24°</td>
<td>None</td>
</tr>
<tr>
<td>6</td>
<td>Partial Shroud</td>
<td>11 14</td>
<td>12 12</td>
<td>12°</td>
<td>Moderate</td>
</tr>
<tr>
<td>7</td>
<td>No Shrouding</td>
<td>12 15</td>
<td>12 12</td>
<td>20°</td>
<td>None</td>
</tr>
<tr>
<td>8</td>
<td>No Shrouding</td>
<td>14 14</td>
<td>15 16</td>
<td>12°</td>
<td>None</td>
</tr>
<tr>
<td>9</td>
<td>Full Shroud</td>
<td>11 24</td>
<td>9 11</td>
<td>12°</td>
<td>Large</td>
</tr>
</tbody>
</table>
vorticity generated within the continuous boundary layers. It is clear that the introduction of shrouding around the bellmouth-intakes to physically suppress the upstream eddy-shedding, will interfere with the stream-line distribution into the intake (Fig. 7e). It is therefore logical to suppose that the type of shrouding offering the maximum eddy suppression and minimum stream-line interference will be the most desireable.

The best results recorded were given by the 'Partial Shrouding' (Fig. 11b), showing the effect of moderate stream-line interference with the suppression of eddies shed from the intake pipe alone (not the bellmouth) at a sump angle of $24^\circ$. It is reasonable to suppose that the 'Mesh Screen' (Fig. 11d) at $\alpha = 24^\circ$, would have given better results by suppressing eddies from both the upstream pipes and bellmouths, and at the same time offering the least stream-line interference.

The 'Full Shrouding and Bottom' at $\alpha = 24^\circ$ would be the likely second, exhibiting the maximum stream-line interference, but which is compensated for by a third boundary layer*, established between the shroud bottom and the floor clearance, and is additional to the two bi-stable boundary layers formed between the sides of all shroud types, and the channel side walls.

*(c.f. Appendix VI)
DISCUSSION

Anemometry

Considerable trouble was experienced initially in obtaining reliable anemometer readings. Small bubbles and dirt particles found to accumulate on the surface of the probe's sensor, affected the heat transfer rate between the sensor and the fluid and caused the readings to become unstable.

The dirt accumulation (mostly fine iron-oxide in suspension) brought about a steady decline in the readings, whereas the bubble formation caused a much faster fall-off, which tended to rise abruptly as soon as the bubbles had grown large enough to be dragged off the sensor into the main stream flow. Graph No. 2 illustrates these effects.

The problem was minimized in the following ways:

DIRT REDUCTION:

a) all ferrous fittings exposed to the test water within the flume were removed and the flume repainted.

b) by frequent changing of water and cleaning down of flume itself. (ultimately limited by the cleanliness of the incoming water supply).

c) by the cleaning of the probe's sensor directly before each reading with a pure solvent (methyl hydrate) and fine brush.

BUBBLE REDUCTION

Bubble formation seems to be an inherent problem associated with re-circulation systems, due to the high levels of dissolved air maintained. The precautions taken were:
To ensure that the test water temperature was \( \sim 65^\circ F \), thereby increasing the water vapour pressure and decreasing the tendency for dissolved air to remain in solution;

To use the probe with the lowest over-heat ratio, and physically vibrating it, so as to shake off the bubbles on the threshold of breaking free, before each reading;

Detergents were added to the test water to reduce the surface tension and subsequent bubble formation on the sensor. Only bubbles with an initial radius greater than a certain threshold value predetermined by the surface tension will grow whilst those with radii smaller than this value, shrink and dissolve back into the fluid. Hence the less the surface tension, the less the forces acting within the fluid on the bubbles and the greater the threshold radius, culminating in more air being re-dissolved and a lesser bubble formation.

Any improvement gained on this account was offset by the detergent's accelerated effect upon the rust content accumulating within the system, which also effected the sensors heat transfer rate and accuracy.

Recommendations for the Future Usage of Hot Film Anemometry in Fluid Flow:

a) Because of the probe's sensitivity to dirt, a cleaner water supply source is desirable. Similarly a smaller more enclosed flume would aid the prevention of environmental dust from entering the system.

b) The use of a conical pencil type probe sensor, offering a reduced leading edge and minimum stagnation point, would decrease the propensity for dirt to accumulate on its surface, by virtue of its more stream-line profile.
c) Because the turbulent frequencies observed were low \((\leq 500 \text{c/s})\), the loss in response due is the use of a conical probe rather than a wedge type, would not affect the results.
d) To facilitate easier \((u')\) readings of the high frequency velocity perturbations, a larger time scale of \(\tau = 0.3\) or \(\tau = 1.0\) seconds in place of \(\tau = 0.1\) seconds is recommended.
e) The T.S.I. Signal Suppressor Module (No.1057) high and low pass frequency filters did not permit sufficient low frequency isolation in the \((2-200)\) c/s range to enable a comprehensive low frequency spectral analysis to be performed.

Future experimentation could well use a more comprehensive set of frequency filters in conjunction with a continuous external recorder.

**Future Model Test Philosophies:**

The ultimate assessment of a flow within a sump, is its effect or lack thereof, upon the pumped efficiency. In the ideal case, it is proposed that future model sump tests should:
a) incorporate direct pump efficiency tests, to be carried out in conjunction with
b) a sump turbulence study using a three dimensional anemometer sensor, to enable both the worst flow conditions and the positions at which they occur to be determined, as well as the Turbulence Intensity values, useful for future test comparisons.
c) the model sump and system should be used in conjunction with pumps of the same specific speed as the prototype, to simulate parameters such as the same bellmouth stream-line distribution, pre-rotation of flow in the intake and the same net positive suction head conditions.
d) a sensitive suction pressure device, mounted adjacent to the intake, would provide additional useful indication of the presence of organized flow instabilities.

e) a series of tests designed to determine a relationship between a combined large and small scale turbulence parameter with the pumped efficiency, might provide a useful basis from which to assess the contractual disputes between manufacturer and client, arising from the differences in performance between factory shop tests and full load site efficiency tests.

At the present moment, there is no reasonable way of assessing them.
SECTION VI: CONCLUSIONS

1) Under some conditions an 'in-line' sump configuration may offer large economies, especially underground. The present study indicates the feasibility of constructing such a sump that would work.

2) The analysis of the results indicated that improvements over the initial poor flow conditions were achieved by the use of various types of bellmouth shroudings, to physically suppress the organization of vorticity shed from the adjacent upstream pumps, and increases in the sump channel angle ($\alpha$) improved the flow conditions within the sump by decreasing the channel flow's deceleration.

3) The order of effectiveness of the different shroud types tried was found to be:

- Partial shrouding at $\alpha = 24^\circ$
- Wire mesh screening at $\alpha = 20^\circ$
- Full shrouding at $\alpha = 24^\circ$
- Full shrouding & bottom at $\alpha = 20^\circ$

When compared to No Shrouding at $\alpha = 24^\circ$.

Since the flow conditions were seen to improve with an increase in the sump angle, the potential order of the shrouding devices tested at $\alpha = 24^\circ$, would likely have been:

- Wire mesh shrouding
- Partial shrouding
- Full shrouding & bottom, and
- Full shrouding.
4) The readings taken using the unidirectional anemometer sensor provided a moderate indication of the turbulence levels within the sump, though more reliable results would have been obtained with a three-dimensional sensor.

5) In general, turbulence intensities \( \frac{u'}{U} \), are a useful standard basis by which to compare the flow conditions within different sumps, though for purposes of these tests, the mainstream velocity fluctuations \( dU \) were used in preference.
BIBLIOGRAPHY


Other Works Consulted


NOMENCLATURE

C - bellmouth to floor clearance
D - bellmouth diameter
\( \bar{d}_p \) - intake pipe diameter
E - general energy expression
F - force
g - viscous dissipation losses
H - total depth of fluid
h - head at any point
K - a constant
k - distance from bottom to any point
L - traverse position of probe from sump wall
N - revolution count
P - pressure at any point
\( P_\infty \) - atmospheric pressure (h_a)
r - radius at any point
S - bellmouth submergence below the water surface
T - surface tension force
t - time
U - main stream flow velocity
\( \overline{U} \) - average of main stream flow velocity
u - general term for velocity at a given point
u' - fluctuating component of velocity at any point
\( \overline{u}' \) - mean fluctuating component of velocity at any point
V_a - axial velocity component
$V_T$ - velocity component at impellor tip

$V_h$ - velocity component at impellor

$V_t$ - whirl or tangential velocity component

$\alpha$ - angle between the two sides of the sump (sump angle)

$\lambda$ - wave length of velocity fluctuation

$\rho$ - density of fluid

$\omega$ - angular velocity of spin at a point

$\mu$ - viscosity

$\phi$ - velocity potential function

$\psi$ - stream function

$\sigma$ - surface tension

The standard notation for cylindrical co-ordinates $x,y,z,V_T,V_0$, and $V_z$ are shown on Fig. 10b.
The Significance of Vortex Theory to Pump Design

Ideally all pump impellor design starts out from the assumptions that the flow from the sump into the pump is:—

a) axial

and

b) uniform in its distribution.

The following brief discussion indicates the necessity for these premises.

1) Free Vortex Theory: It can be shown that a fluid flowing with a free rotation* may be represented by the Euler's equations in a reduced form:

$$\frac{dP}{f} + \frac{V_t dV_t}{g} = 0$$

Since the flow is considered to be in radial equilibrium, the centrifugal forces due to angular rotation are exactly balanced by the pressure across the field, such that $dP = \frac{V_t^2}{g} \frac{dr}{r}$.

i.e. at any point pressure $\rightarrow$ Centrifugal force $\frac{MV^2}{r}$

(element of fluid in a rotating field.)

By combining these two expressions, we obtain: $\frac{dV_t}{V_t} + \frac{dr}{r} = 0$

or $V_t \cdot r = \text{constant.}$

* i.e. unacted upon by any imposed forces.
Similarly, the introduction of an axial velocity component does not alter the relationship. i.e. The Euler equation \(+\) additional component becomes: 
\[
\frac{dp}{\rho} + \frac{V_a}{g} \frac{dv_a}{dt} + \frac{V_a}{g} \frac{dv_a}{g} = 0
\]
which, when combined with the pressure gradient \(\frac{dp}{\rho}\) and integrated, still reduces to \(V = \) constant where \(V_a = 0\) is a trivial solution and therefore \(d(V_a) = 0\). This simple condition is applied directly to axial flow impellor design, where the axial flow velocity distribution remains constant with radius, even when superimposed upon the free rotational flow pattern within the impellor. By maintaining radial equilibrium, all secondary flows are eliminated and the fluid viscous shear losses are kept to a minimum, giving a maximum fluid efficiency condition with:

\[
V_a = \text{constant - root to blade tip}
\]
and
\[
V_e = \text{constant - root to blade tip}
\]
To accommodate the linear variation of the peripheral velocity with radius and to keep the impellor through-flow in a parallel cylindrical streamline form for maximum efficiency, the impellor blade angles are varied at both the inlet and outlet edges. (cf. Figs. A & B) This tends to give a twist to each blade along its length, making it desirable to keep the blade outlet angles small so as to minimize the fluid energy losses through the impellor.

Blade profile design has progressed through several stages of development, and for a time the use of aerofoil blade profile data, with very small outlet angles \((2-4^\circ)\) was practised*. Because of the small

* Fluid enters the impellor inlet, (centre of the forced vortex) where it is whirled around and ejected at the periphery, at a much higher velocity, which is reconverted into pressure energy by the radial diffuser.
PRESSURE AND VELOCITY DISTRIBUTION THROUGH AN AXIAL FLOW PUMP WITH DIFFUSER.

Pressure exchange diagram:

Blade velocity diagrams:

\[ U_T = \omega r_T \text{ peripheral TIP velocity} \]
\[ U_H = \omega r_H \text{ peripheral HUB velocity} \]
\[ V_T > V_H \]
\[ V_A = \text{constant} \]
VELOCITY AND PRESSURE DISTRIBUTION THROUGH AN AXIAL FLOW TURBINE WITH INLET GUIDE VANES \textsuperscript{(12)}

![Diagram of axial flow turbine with indicated stations and pressure/velocity distribution graphs.]

Station A
- Static pressure
- Velocity head
- Total pressure

Station B
- Static pressure
- Velocity head
- Total pressure

Station C
- Static pressure
- Velocity head
- Total pressure

Flow direction indicated by arrows from root to tip.
outlet angles, the fluid exit velocities at the blade tip were often greater than those at the hub, necessitating the use of a diffuser with straightening blades, for the purposes of a uniform pressure recovery. Present-day axial flow pump design, more frequently uses the mathematical conformal transformational approach to geometric blade-profile design, directly incorporating the free vortex theory. Axial flow turbines invariably have inlet guide vanes, which induce the free vortex distribution in the flow before entry into the propellor, and consequently dispense with the need for diffusers.

Forced Vortex Application: Forced vortex theory is normally associated with the design of mixed and radial flow machines, because of the velocity increase with radius within the vortex region (cf. Fig. 2.). \( V/r = \text{Constant} \).

However, a forced vortex with an axial flow superimposed has frequently been used as the design basis for axial flow machinery requiring no inlet guide vanes. (13) The Euler head may still be generated by a forced vortex distribution, assuming a uniform and axial flow at inlet the impellor exists. However, the impellor must be designed for a constant pitch along its radius to maintain a constant axial velocity, and increasing pitch along the length of the blade from the leading edge.

Where the pitch

\[
P = \frac{\pi D \tan B}{V_a}
\]

Pitch/sec - \( P_s = \frac{ND \tan B}{V_a} \)

and \( D = \text{impellor dia} \)

\( B = \text{impellor vane inlet angle} \)

\( V_a = \text{axial velocity of fluid} \)

\( N = \text{rev./sec.} \)

\( P = \text{pitch as defined by Stepanoff} \). (2)
The pitch at the inlet edge determines the normal axial velocity $V_a$, and the ratio of the pitch/sec. at outlet is a measure of the impelling action of the impellor, known as impelling ratio $\frac{P_s}{V_a}$.

There are a number of distinctive advantages in using the forced vortex theory in axial flow machines as discussed by Stepanoff\(^{(12)}\), though present day practises give higher efficiencies using the aerofoil or conformal transformation approaches.
Boundary Layer Momentum Exchange and Vorticity

If a sump's walls are of reasonable continuity and sufficient length, a boundary layer will be established within the passing flow. Within the limits of this boundary layer, a continuous exchange of momentum occurs and vorticity is generated. Adjacent to the walls there is a small loss of momentum within the fluid, due to the action of viscous drag. Extraneous momentum from upstream turbulence is interchanged with the boundary layer and the resultant angular momentum is characterized by the vorticity generated, whose strength is a function of the main stream flow velocity. (U)

By definition, Circulation \( \Gamma = \oint \mathbf{V} \cdot d\mathbf{s} \)

which, for the element \( \Gamma_{ABC} = v_{OA} \cdot x + v_{AB} \cdot y + v_{BC} \cdot x + v_{CO} \cdot y \)

but \( v_{OA} \) along the boundary = 0, and for laminar flow \( v_{AB} = v_{CO} = 0 \)

\( \because \Gamma = 0 \cdot x + 0 \cdot y - U \cdot x - 0 \cdot y = U \cdot x \)

The average vorticity \( \xi = \frac{\Gamma}{A} = \frac{-ux}{xy} = \frac{-U}{Y} \)
Thus, we can see that the vorticity generated within the boundary layer is a function of the mainstream velocity, and remains independent of the boundary layers' velocity distribution.
Derivation of Hattersley's Swirl Parameters (cf. Fig. 5)

Hattersley\(^{(7)}\) maintained that the vorticity generated within the boundary layer was a function of the velocity distribution, which in the case of a turbulent boundary layer may be shown to be related to the fluid's average shear stress at the walls, and be conveniently expressed in terms of the mean channel velocity (\(\overline{V}\)) and the friction factor (\(f\)), such that:

\[
V = \overline{V} \sqrt{f}
\]

His experimental observations indicated swirl to be sensitive to viscosity changes. Thus, by combining all three parameters into a form of the Reynolds number, with the dimension of length (\(d_e\)) expressed in terms of the hydraulic radius of the channel, he arrived at the expression:

\[
\frac{8 \, V \cdot d_e}{\overline{V}} = R \sqrt{f}
\]

where the mean channel velocity \(\overline{V} = \frac{Q}{A} \text{ ft}^3/\text{sec}\)

\(f\) = COEF. of friction for a circular pipe at Reynolds Nos. \(<10^5\).
\(\nu\) = kinematic viscosity of water
\(Q\) = channel flow
\(A\) = channel cross sectional area
\(d_e\) = equivalent dia. of the channel's cross section

This parameter was thought to aptly combine all the features influencing the formation of swirl, which was measured by a series of freely pivoted meridional vanes (vortometer). The resultant vortometer revolution count (\(N\)), converted to radians/sec gave the "indicated swirl parameter" - \(\tan \alpha\).
Tan \( \alpha \) was the ratio of the linear tangential velocity at the vortometer vane tip, to the average suction pipe axial velocity \( (V_a) \); such that:

\[
\tan \alpha = \frac{\pi N d_p}{V_a}
\]

where \( d_p \) = suction pipe diameter

Fig. 5 then, is a plot of the relationship \( \tan \alpha \) against \( R \sqrt{f} \).
A Typical List of Rules* for Good Design in Sump and Approach Channel

**Approach Channel**

1. Approach channel's velocity should be within the range 1-4 ft/sec (pref. 3 ft/sec).
2. The channel's walls should be straight for a significant distance (say 50 ft. or more).
3. The channel's intake should be submerged, with the roof of the intake sloping upwards so as to leave the approach to the pump with a free surface (say 30 ft.).
4. The part of the approach channel with a free water surface should be roofed over, to prevent wind disturbance upon the water.
5. The final approach to the suction pit should be made with convergent walls, and tangential to the curved or elliptical wall encompassing the rear of the pit.

**Sump**

6. An optimum flow should be established within the approach channel.
7. If approach is straight, the pump's axis should be disposed symmetrically on the centre line of the approach flow.
8. If approach to sump is asymmetrical, the flow should be divided into two streams, which are then controlled by baffles to achieve a balanced flow into the bellmouth.
9. To inhibit vortex formations upstream of the pump, horizontal splitters located at a minimum height approximately one bellmouth dia. above the floor may be used.
10. To inhibit vortex formation at the sides and rear of the bellmouth
- artificially roughen the rear wall of the sump to produce sufficient turbulence to break up the formation
or
- carry out model tests to produce optimum conditions.

* Proposed by Young (10), based on work done by (3) (4) & (7).
'HOMOGENEOUS TURBULENCE'

Early investigations indicated that homogeneous turbulence fulfilled most of the requirements for isotropy, where isotropic flow has been defined* as one in which the statistical parameters remained invariant to reflection and rotation of the co-ordinate axes. However, the decay of turbulence in a stream wise direction actually precludes a completely isotropic state, but if the analysis is restricted to narrow lateral planes this difficulty may be ignored.

Consider a thin lateral plane within a turbulent fluid flow. (Fig. C). The fluid particles are not retained in layers, but more in a random hetro genius fashion, colliding with one another, causing a mixing of the layers and a rapid irregular pulsation of the velocity.

Thus, the instantaneous velocity \( U' \) may be considered to be composed of the vectorial sum of the temporal mean velocity \( U \), and the pulsating components \( V_x \) and \( V_y \) which are both functions of time.

The R.M.S. of the \((V_x)\) and \((V_y)\) values give a measure of the violence of the turbulence fluctuations, and the magnitude of the departure of \((U')\) from \((U)\) is widely known as the Intensity of Turbulence.

The mean time interval between the sign reversals of \((V_x)\) and \((V_y)\) is a measure of the Scale of Turbulence \((\overline{\lambda})\).

When the fluctuating velocity components \(V_x = V_y\) and \(-V_x = -V_y\) the turbulence is said to be homogeneous. Since for practical purposes nearly isotropic turbulence may be assumed, the turbulence intensity may be written:

\[
T.I. = \frac{\sqrt{(U')^2}}{U}
\]
THE NEUTRALIZATION OF VORTICITY WITHIN A BI-STABLE BOUNDARY LAYER

All vorticity is generated within the boundary layer. The direction of spin of rotating diffuse vortex elements \( (+\omega) \) at the boundary surface is dependant upon the position of the boundary surface to the main stream flow direction:

When two fixed boundary surfaces are in the same proximity and the main stream flow remains uniformly equi-distance between them, a bi-stable boundary layer within the flow is formed:

The two sets of opposite directional spinning diffuse vortex elements cancel each other out, so that \( \omega = 0 \).
PHOTOGRAPHS

2- Test Channel

3- Test Channel

4- Pump Arrangement
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