SUBMERGED MULTIPOOL DIFFUSERS
IN SHALLOW WATER WITH CURRENT

by

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(1970)

Submitted in partial fulfillment
of the requirements for the degree of
Master of Science
at the
Massachusetts Institute of Technology
May, 1972

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JUL 12 1972
ABSTRACT

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Submitted to the Department of Civil Engineering on May 12, 1972 in partial fulfillment of the requirements for the degree of Master of Science in Civil Engineering.

Increased electric power production has brought with it strict standards for thermal discharges. Submerged jet diffusers are required to meet these standards in shallow coastal waters.

Downstream dilution from a shallow water diffuser in a current is studied under the assumptions of no stratification, no bottom friction, and boundaries at infinity. Momentum and energy equations are used to predict dilution for the three cases of nozzles aligned with a current, nozzles aligned against the current, and nozzles alternated. The approach is similar to that used in classical propeller theory. Results are verified by an experimental model study.

Dilution depends on crossflow momentum and induced flow momentum. When these two act together, dilution is maximized, and when they are opposed, dilution is minimized. The design of prototype shallow water diffusers is discussed using these principles.

Thesis Supervisor: Donald R. F. Harleman
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ACKNOWLEDGMENT

The experimental portion of this study was performed for Stone and Webster Engineering Corporation of Boston under DSR 72454.

I extend thanks to my advisor, Dr. D. R. F. Harleman for his guidance throughout this work; to Mr. Gerhard Jirka and Dr. Keith Stolzenbach for their plentiful advice; and to Miss Justine Lynge who helped with the typing.
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I INTRODUCTION

Thermal pollution is a common subject when discussing electric power production. Nearly all of the new power plants being built are designed on a thermal cycle which burns fuel, either fossil fuels such as coal or oil, or fissionable nuclear fuels, to produce steam for driving turbines. The steam is then cooled in condensers before being recycled through the boiler. The efficiency of such cycles is controlled by the maximum operating temperature in the cycle and is usually in the range of 30-40%. Thus for every Kilowatt-hour of electricity, the equivalent of two Kilowatt-hours of thermal energy is transferred through the condenser into the environment.

Conventional sinks for this heat are nearby water bodies, hopefully oceans with swift currents, uninteresting marine biology, and sparse, uncomplaining population, but just as likely they are shallow bays, lakes or rivers with limited cold water supplies and nearby populations for whom "thermal pollution" is one more reason to avoid building a power plant nearby.

The environmental effects of increasing water temperature in a given region are not always known. Individual states, however, are setting temperature standards, and eventually it is expected that these will conform to those suggested nationally by the Environmental Protection Agency. These standards usually limit the maximum temperature rise in a given water body and often include an initial mixing zone in which dilution must take place down to the prescribed temperature. Standards may have different limits for rivers, lakes, and ocean and estuarine areas.
In addition, every site has its own conditions of current, ambient stratification, geometrical boundaries, etc., which influence the spreading of hot water often making the design of an outfall unique for a particular area.

Significant parameters in comparing discharge schemes are the temperature rise above a reference, $\Delta T$, and dilution at some defined region. These are reciprocal and are defined by

$$\Delta T = T - T_r$$

$$S = \frac{T_o - T_r}{T - T_r} \quad \text{or} \quad S = \frac{\Delta T}{\Delta T_0}$$ (1.1)

where $T_o$ is the hot water temperature, $T_r$ is a reference water temperature and $T$ is the temperature at the point of measurement. At some distance downstream from the outfall the temperature rise may achieve a stable minimum and the dilution a corresponding maximum. Further temperature change takes place only through the slow process of atmospheric and dispersive heat transfer, and hence at this point a representative average dilution may be defined.

For engineering application, the challenge is to maximize $S$ or at least to achieve an acceptable minimum value. One way to do this is to discharge the hot water through a high velocity jet which through turbulence will entrain cold water from its sides to achieve rapid dilution. Better yet is to divide the flow into as many jets as possible, thus in-
creasing the surface area for entrainment, and to locate the jets near the bottom obtaining additional head in the form of buoyant potential energy. Such a multiport diffuser is particularly useful in shallow water where the available water for cooling is limited and where standards prescribe a small mixing zone necessitating rapid dilution.

The Shoreham Nuclear Power Station on Long Island described in reference (7) is a practical example of a design within such limitations. New York state "summer" standards prohibit surface temperature rises in the Sound of over 1.5°F outside of a mixing zone of about 300,000 square feet. In order to dispose of the projected 1280 cfs. of condenser cooling water raised to 19.7°F above ambient, a 3800 foot diffuser with 65 alternating jets was designed to extend offshore and to discharge into waters of approximately fifteen feet in depth.

Hydraulic model studies from which the above design was picked were performed at M.I.T.'s Parsons Laboratory for Water Resources and Hydrodynamics (2) in the spring of 1971. In the preliminary stages, a range of diffuser orientations were studied to decide upon the best configuration for Shoreham, and afterwards the model was used to test specific orientations for diffusers in general. The experimental data for this thesis is drawn from these two sources. Experiments are then correlated with a theory which predicts a well-mixed dilution as a function of water depth, cross current speed and direction, and jet diameter, spacing and velocity.
II BACKGROUND

A. General

The objective of a jet diffuser is to reduce the concentration (of temperature) through rapid mixing of the discharge water. This dilution is a function of jet parameters and ambient flow conditions. This chapter categorizes some of the possible flow conditions available with shallow water diffusers, and predicts the dilution formula for some simple cases. The definition of dilution will be given below. Several of the cases have been studied at M.I.T., and one is analyzed in detail in this thesis. The others are included for generality and as a format upon which further research could be undertaken.

As a starting point, consider the diffuser within the rectangular basin of uniform depth, \(h_r\), as shown in Figure 2.1. Warm water exits from nozzles parallel to the channel bottom and perpendicular to the diffuser row. In addition there may be an ambient crossflow of velocity \(u_r\) and volumetric flow \(Q_r\) flowing from left to right. A shallow water constraint dictates that no thermal stratification should occur. This condition is satisfied by favorable choice of the ratio of water depth to jet diameter, \(h_r/D_0\), and densimetric Froude number (to be defined later).

We would like to know the well mixed downstream temperature \(T_m\); we seek a formula which will give \(T_m\) as a function of \(T_r\), the jet parameters, and the flow characteristics. \(T_r\) may or may not be the temperature of the entrained water, \(T_a\). If any of the diffuser discharge returns to the diffuser and is re-entrained, \(T_a > T_r\). Note the distinction be-
Figure 2.1 A Typical Shallow Water Diffuser
tween re-entrainment at the diffuser line, and recirculation of warmed water at the condenser intake. We presume the latter does not occur. Assume then that we want $T_m$ only as a function of $T_a$. In analogy with

Equation (1.1) we can write

$$S^* = \frac{T_o - T_a}{T_m - T_a}$$  \hspace{1cm} (2.1)

where $S^* = S$ if $T_a = T_r$. Given $T_o$ and $T_a$, to solve for $S^*$ is to solve for $T_m$. But clearly $T_m$ is a function of $T_a$ and the quantity of flow $Q_m$ drawn through the diffuser by the ratio of flows,

$$T_m = \frac{Q_m T_a + T_o Q_o}{Q_o + Q_m}$$  \hspace{1cm} (2.2)

With algebra this yields

$$\frac{T_o - T_a}{T_m - T_a} = \frac{Q_m + Q_o}{Q_o}$$

Since

$$S^* = \frac{T_o - T_a}{T_m - T_a}$$

it follows that

$$S^* = \frac{Q_m + Q_o}{Q_o}$$  \hspace{1cm} (2.3)

Equation (2.1) is a temperature dilution, while (2.3) is a volumetric dilution. The latter can be used to calculate temperatures in the former. $Q_m$ is derived from two sources: ambient cross flow and the flow
induced by the individual jets' momentum. If a diffuser were standing in stagnant water, all of its dilution water would have to be induced from the jets. This extreme is called induced flow mixing. On the other extreme is the case where the diffuser has little or no net momentum in comparison with the momentum of the crossflow which passes over it. This extreme is labelled purely crossflow mixing. For in between cases dilution will be derived from both induced flow and crossflow mixing, with the relative magnitudes depending on the crossflow Q_r and the jets' momentum. This chapter's analysis begins with the no crossflow case (pure induced flow mixing), then considers the combination of crossflow and induced flow mixing, and finally considers the extreme of pure crossflow mixing.

B. The No Crossflow Case (Induced Flow Mixing)

With no crossflow, the situation is depicted by Figure 2.2, and in accordance with (2.2) the variables of interest are T_a and Q_m (or S*).

One way to look at Q_m is to consider the individual jets comprising the diffuser. If the volumetric flow induced by one jet can be determined, Q_m can be expressed as the sum of the flow from all jets. Fig. 2.3 shows a single turbulent jet with initial velocity u_o, crosssectional area a_o, flow rate Q_o/n = u_o a_o, momentum p u_o^2 a_o, and temperature rise T_o - T_r = ΔT_o. As the jet leaves the nozzle it spreads due to turbulent entrainment of side water. If s represents the coordinate along the jet centerline, then the outer edge grows as b(s), the crosssectional area as a(s), and the total flux within a(s) as Q(s). It is Q(s) which we are
Figure 2.3 A Single Submerged Jet
after. The centerline trajectory is a function of the initial densimetric Froude number,

\[
F_d = \frac{u_o}{\sqrt{\frac{\Delta \rho}{\rho} g D_o}} \quad (2-4)
\]

where \(D_o\) is the nozzle diameter and \(\frac{\Delta \rho}{\rho}\) is a density anomaly due to buoyancy,

\[
\frac{\Delta \rho}{\rho} = \frac{\rho_a - \rho_o}{\rho_a} = \beta \frac{T_o - T_a}{\rho_a}
\]

where \(\rho_o\) is the density of the hot water, \(\rho_a\) is the density of ambient water, and \(\beta\) is a coefficient of thermal expansion. \(F_d\) represents the ratio between momentum and buoyancy forces, and for high \(F_d\) in shallow water, the trajectory is nearly horizontal and the coordinate \(s\) may be approximated by the longitudinal coordinate \(x\).

By solving the momentum and continuity equations, and assuming similarity profiles for the horizontal velocity and temperature distributions (Fig. 2.3), \(u = u_c(x)f(\eta)\), \(\Delta T \equiv \Delta T_c(x)g(\eta)\), the following relations may be obtained, valid after a short distance from the nozzle.

\[
u_c(x) \sim x^{-1} \quad (2.5.a)
\]
\[\Delta T_c(x) \sim x^{-1} \quad (2.5.b)
\]
\[b(x) \sim x \quad (2.5.c)
\]
\[a(x) \sim x^2 \quad (2.5.d)
\]
\[Q(x) \sim x \quad (2.5.e)
\]

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Centerline dilution, $\Delta T_o/\Delta T_x(x)$, and an averaged dilution $Q(x)/Q_o$, both increase as $x$, or combining with (2.5d),

$$S^* \sim a(x)^{1/2}$$
or

$$S^* \sqrt{\frac{a(x)}{a_o}}$$

when normalized with respect to nozzle area

Now consider the jet at the bottom of shallow water with submergence $h_r - h_o = h_{r'}$, and alongside $n$ other jets of uniform spacing $b=L/n$. As it expands, the jet will run into the free surface and the bottom, and spread into adjacent jets. If $h_r$ and $b/2$ are nearly equal, these two will occur simultaneously and $a(x)$ is approximated by $h_r b$ or simply the area "occupied" by each jet in the diffuser, $A/n$ where $A=Lh_r$. The volumetric dilution, $S^*$ is thus

$$S^* \sim \sqrt{\frac{A/n}{a_o}}$$

and if all jets contribute equally (2.7) will apply for the whole diffuser,

$$S^* \sim \sqrt{\frac{A}{na_o}}$$

We have assumed that the submergence depth is about one half of the spacing so that the jets will surface and interfere with adjacent jets simultaneously. This, however, may not be the best design. Consider, for instance, wider spacing where $b>2h_r$. Jets will hit the surface before they interfere with each other. From the point of surfacing, spreading will be limited to the lateral direction as if the jet were
two-dimensional. The relations in (2.5) can be rederived for the present case from the point of surfacing

\[ u_c(x) \sim x^{-1/2} \]  
\[ \Delta T_c(x) \sim x^{-1/2} \]  
\[ b(x) \sim x \]  
\[ a(x) \sim x \]  
\[ Q(x) \sim x^{1/2} \]

(2.8.a)  
(2.8.b)  
(2.8.c)  
(2.8.d)  
(2.8.e)

where instead of being circular, \( a(x) \) now describes a rectangular region of height \( h_x \) and width \( 2b(x) \). Equations (2.5.e) and (2.6) still hold however with only a change in the constant of proportionality. Thus dilution goes up with increased spacing. There will be a point of diminishing return, though as the dilution attainable from wider spacing approaches that for isolated jets. This topic is discussed by Cederwall (1) and Larson and Hecker (3). Unfortunately though, their analysis does not include the shallow water case.

We would expect in summary, that for conservative spacing, dilution with no cross flow should behave in accordance with equation (2.7). An alternate approach to the no cross flow case predicts a coefficient in (2.7) by treating the diffuser as a laterally uniform source of momentum with no consideration for the individual jets. This approach is described below.

Harleman, Jirka and Stolzenbach combined an analytical and experimental study of the no cross flow case in Phase I of their study (2). Figure 2.4 shows a model set up with three jets inside channel walls with
a) Model Set Up

b) Energy Considerations

Figure 2.4 Phase I Study after Harleman et al (2)
the whole apparatus inside basin walls. The channel walls prescribe the path of the induced flow which was allowed to run until steady state velocities were achieved and velocities and temperatures were laterally and vertically homogeneous in the upstream and downstream portions of the channel. At steady state it is presumed that the induced flow undergoes no net change in momentum in making a complete circuit. Therefore, the momentum which is given the flow, \( pu_o^2 a_o \), is converted into a head

\[
\Delta H = \frac{u_o^2 a_o}{gA/n}
\]  

(2.9)

and is entirely dissipated as losses. For this study, bottom friction was considered negligible and only the entrance and exit losses associated with rounding the channel walls were considered, and were assumed proportional to \( K_1 u_U^2/2g \) and \( K_2 u_D^2/2g \) respectively where \( u_U \) and \( u_D \) are upstream and downstream velocities in the channel.

Invoking conservation of momentum between sections U and D and Bernoulli's equation between D and U (outside of the channel walls), the following relationship was achieved:

\[
S^* = \sqrt{\frac{2bh}{\frac{Q_o + Q_U}{4}(K_1 + K_2)}}
\]

where \( S^* = \frac{Q_D}{Q_o} = \frac{Q_U}{Q_o} \). Using \( K_1 = .5 \) and \( K_2 = 1.0 \), this becomes

\[
S^* = \sqrt{\frac{4}{3}} \sqrt{\frac{A/n}{a_o}}
\]  

(2.10)
which has the same form as Equation (2.7).

No effort was made to evaluate warm water re-entrainment. Experimental values of dilution were based on instantaneous ambient temperature $T_a$, just upstream from the jets, and as $T_a$ built up with time, so did $T_m$ leaving $S^*$ constant.

In cases of no cross flow, re-entrainment is indeed significant but there is no easy way to describe it analytically. From Figure 2.2 and Figure 2.4, it is clear that the diffuser in stagnant water will set up a basin wide circulation and all water will eventually be recycled at a temperature $T_a$ at the point of entrainment. $T_a$ depends on the atmospheric heat loss rate and the path length of circulation, with the basin side and end walls playing a significant role in reducing this path length, as compared to the wide open case.

C. Combined Crossflow and Induced Flow

When a cross flow is added, the diffuser is able to draw some, or all of its $Q_m$ from the ambient flow $Q_r$. Two hypothetical cases are shown in Figure 2.5. For mild cross flows, (Case A) insufficient water is available from $Q_r$ and portions of the mixed waters are re-entrained to make up the difference. The temperature of entraining water $T_a$, is therefore non-uniform and greater than $T_r$. For larger cross flows (Case B) $Q_r > Q_m$, no re-entrainment will take place and $T_a = T_r$.

Figure 2.5.b pictorially divides the mixing flow $Q_m$ into that which is supplied by the cross flow, $Q_c$, and that which is induced by the
a) Mild Crossflow

b) Stronger Crossflow

Figure 2.5 Diffuser with Crossflow
diffuser, \( Q_1 \). The two cases are

\[
Q_m = Q_c + Q_i > Q_r \quad \text{Case A} \quad (2.11.a)
\]

\[
Q_m = Q_c + Q_i < Q_r \quad \text{Case B} \quad (2.11.b)
\]

\( Q_c \) can be stated in terms of the upstream discharge or velocity by

\[
Q_c = \frac{L}{Y} Q_r = u_r L h_r \quad (2.12)
\]

Narrow channel walls as would occur in rivers and in experimental models enhance the chance of re-entrainment by limiting \( Q_r \). In order to use an experimental model to model a prototype diffuser in relatively open waters, the ratio \( L/Y \) need not be followed exactly, but it should be small enough to avoid shifting from the regime of B) to the regime of A).

In Phase II of their study, (2), Harleman et al extended their theory to cover the case shown in Fig. 2.5.b, and included the case where the direction of the nozzles was reversed, and pointed into the cross flow. Figure 2.6 shows hypothetical streamlines for their two cases.

Again using the momentum and Bernoulli equations (with the inclusion of bottom friction this time) and assuming \( S^* >> 1 \) the following results were obtained,

\[
S^* = \sqrt{\frac{Q_c}{Q_0} \left( \frac{2 bh_r}{2bh_r + \frac{\pi D^2}{4} (K_b - K_2)} \right)} \quad (2.13)
\]
a) Nozzles with Crossflow

b) Nozzles Against Crossflow

Figure 2.6 Phase II Study after Harleman et al (2)
where $K_b$ and $K_2$ are bottom and exit loss coefficients and the plus and minus refer to cases of jets with and against cross flow respectively. When $K_b$ was set equal to 4 and $K_2 = 1$ as before, good agreement with data was achieved and the equation reduced to

$$S^* = \sqrt{\left( \frac{Q_c}{Q_o} \right)^2 + \frac{2}{5} \left( \frac{A/n}{a_o} \right)} \tag{2.14.a}$$

$$S^* = \sqrt{\left( \frac{Q_c}{Q_o} \right)^2 - \frac{2}{3} \left( \frac{A/n}{a_o} \right)} \tag{2.14.b}$$

for the two cases. The effect of the cross flow dilution and the induced dilution are seen in the two terms $(Q_c/Q_o)^2$ and $(A/n)/a_o$, respectively. Equation (2.14.a) reduces to the form of (2.10) for the limit of $Q_c$ approaching zero. (Only the coefficient has changed.)

Chapter III of this thesis presents an alternative formulation for $S^*$ for these two cases as well as suggestions for the general case of an inclined diffuser not perpendicular to the cross flow. The basic difference is that Chapter III's theory assumes an infinite basin and no friction. The two parameters $(Q_c/Q_o)^2$ and $(A/n)/a_o$ representing cross flow and induced flow dilution appear in a slightly different form.

D. The Pure Cross Flow Case (No Induced Mixing)

This case can occur under any of three conditions:

a. The cross flow is so high that $(Q_c/Q_o)^2 >> (A/n)/a_o$
a) Infinite Crossflow Velocity

b) Full Width Diffuser

Figure 2.7 Diffusers with only Crossflow Mixing
b. The diffuser nozzles are alternated, pointed vertically or in other ways configured so as to provide no net momentum in the longitudinal direction. The term $A/n_{\alpha}$ effectively drops to zero.

c. The diffuser extends across the full width of the channel prohibiting lateral entrainment. Although the flow will be accelerated through the diffuser, continuity of flow requires that the flow downstream equal the flow upstream plus the diffuser flow, and the fluid will be decelerated by the adverse pressure gradient formed by the increased water level downstream.

Dilution in all three cases reduces to the simple equation:

$$S^* = \frac{Q_c + Q_o}{Q_o}$$

or

$$S^* = \frac{Q_c}{Q_o}$$

for $S^* \gg 1$

E. Summary

Although the cases above are not completely general they are relevant in a significant number of design cases, and they represent the extent of existing theory. The only unresolved parameter in these is the quantity of re-entrainment, or the effective ambient temperature $T_a$, in the case where the calculated $Q_m$ is more than the cross flow $Q_r$. 

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More general cases include unsteady cross flows, arbitrary diffuser orientation and irregular basin boundaries.

Unsteady cross flows can occur in coastal and estuary regions under periodic tidal variation or in reservoirs controlled by unsteady dam releases. Were it not for re-entrainment under a tidal reversal, unsteady behavior could be approximated by a step-wise linear cross flow curve, but when the diffuser is liable to be re-entraining its own water the usual recourse is experimental models.

The case of a diffuser oriented obliquely to an oncoming cross flow is analyzed in Chapter III under very relaxed assumptions. For substantial cross flows the diffuser flow will be strongly deflected. Certain orientation angles produce unstable flow situations adding to the complication of the re-entrainment problem. An adequate answer to this case requires knowledge of the deflecting force hitting the diffuser discharge. Fortunately the angle for maximum $S^*$ (see Chapter III) is $\beta = 0$ which is the perpendicular case discussed in this chapter.

The spacing between the basin walls has already been reasoned to affect dilution, and certainly irregular prototype boundaries will influence the circulation pattern in the far field, thus affecting the path length of re-entrainment and the averaged ambient temperature $T_a$. These effects can best be investigated experimentally.
III THEORY

A. Problem Statement

This chapter's theory treats the combined crossflow and induced flow case and again solves for the volumetric dilution \( S^* \). Figure 2.5 shows special examples where the diffuser is perpendicular to the crossflow; these are used as the starting point for the theory. The major difference between this theory and that of Harleman, et al (2) is the absence of any friction factors for matching theory to experiment, and as seen in the following section, friction is small.

Figure 3.1 shows the general problem. A diffuser of length \( L \) in shallow water of depth \( h_r \) discharges \( Q_o \) through \( n \) identical turbulent jets of spacing \( b = L/n \). Nozzle crosssection area is \( a_o \), and jets have uniform initial velocity \( u_o \) and momentum \( \rho u_o^2 a_o \). Jets are located near the bottom parallel to the \( x \)-axis at an angle \( \alpha \) with respect to the bottom. \( 0^\circ \leq \alpha \leq 180^\circ \). An ambient current has uniform upstream velocity \( u_r \) and is oriented at an angle \( \beta \) relative to the \( x \)-axis. \( 0^\circ \leq \beta \leq 180^\circ \) and \( 0^\circ \leq \beta < 90^\circ \) cover all cases; outside of this range, the solution may be found by symmetry. For example \( \beta = 120^\circ \) and \( \alpha = 0^\circ \) is equivalent to \( \beta = 60^\circ \) and \( \alpha = 180^\circ \). In discussing tidal currents, however, it will be convenient to consider \( 0^\circ \leq \beta < 180^\circ \).

The crossflow rate \( Q_c \) is given in Chapter II by

\[
Q_c = u_r L h_r = u_r A \tag{2.12}
\]

where \( A \) is the total "area" of the diffuser. The component of crossflow
passing over the dormant diffuser is \( Q \cos \beta \).

The basin walls define the channel and were of course present in the experimental model; however it is assumed that they are at such a distance from the diffuser that they do not disturb the flow and boundary conditions specify uniform flow. Although re-entrainment is not prohibited, this theory only predicts the quantity \( Q_m \) flowing through the diffuser for use in equation 2.3,

\[
S^* = \frac{Q_m + Q_0}{Q_0} = \frac{Q_m}{Q_0} + 1
\]  

(3.1)

and is not sensitive to differences between \( T_a \) and any upstream reference temperature \( T_r \).

It is assumed, as in (2), that there is no discontinuous velocity increase across the diffuser and that jet momentum is converted into a localized pressure rise \( \Delta H \) which is shortly converted into momentum by invoking the momentum equation across the diffuser (sections 2-3 in Figure 3.2a),

\[
\Delta H = \frac{u_o^2 a_o}{gA/n}
\]  

(2.9)

Reference (2) equates \( \Delta H \) to frictional and minor head losses to determine velocities. The next section shows frictional forces to be small allowing a different approach.

B. Navier Stokes Equations

Assume the nozzles are horizontal--\( \alpha = 0^\circ \) or \( 180^\circ \). The continuity and \( x \) and \( y \) momentum equations governing steady state flow
\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{3.2}
\]
\[
u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -g \frac{\partial h}{\partial x} + \frac{2}{\partial z^2}
u \frac{\partial u}{\partial z} \tag{3.3}
\]

where it is assumed that the only contribution from the vertical (z) dimension is the bottom stress, and velocities in the z direction are negligible. \(\epsilon\) is a kinematic eddy viscosity used to express the turbulent shear stress. Boundary conditions at infinity, or the walls, are all those of undisturbed uniform flow, and at the diffuser, \(\Delta H\) is specified by (2.9), and \(v\) is equal to zero at \(y=0\). However, no "boundary condition" at the diffuser is known for \(u\), for if it were the dilution could be calculated directly from equation 3.4.

The momentum equations suggest a tradeoff among inertia, pressure and eddy viscosity forces. We know that pressure is important for we have hypothesized that jet momentum is initially manifested as a pressure rise. By scaling the equations, the relative importance of inertia and eddy viscosity can be determined. The following dimensionless variables are used:

\[
x' = \frac{x}{L} \quad y' = \frac{y}{L} \quad z' = \frac{z}{h_r} \tag{3.4}
\]
\[
u' = \frac{u}{U_r} \quad \frac{v}{v' = \frac{v}{U_r} \quad \frac{dh'}{dh} = \frac{dh}{\Delta H}
\]

where \(U_r\) is a characteristic velocity proportional to the velocity of a fully developed single jet,
\[ U_r = u_a \sqrt{\frac{A_0}{A/n}} \]  

(3.5)

Assuming all of the primed quantities are \( O(1) \), either momentum equation yields

\[ \frac{U_r^2}{L} = -g \frac{\Delta H}{L} + \varepsilon \frac{U_r}{r} \]

A "Reynolds" number \( R_d \) is formed to determine whether inertial or viscous forces balance pressure.

\[ R_d = \frac{U_r h_r^2}{\varepsilon L} \]  

(3.6)

Prototype values of \( u_r, h_r, L, \) and \( \varepsilon \) are typically 1 ft/sec, 20 feet, 2000 feet, and \( 10^{-2} - 10^{-4} \) ft \( \cdot \) sec, respectively, giving a value of \( R_d \),

\[ 20 < R_d < 2000 \]

Except at large distances from the diffuser where the velocities are small, and \( \frac{X}{L} >> 1 \), eddy viscosity can be considered small and dropped from the equation.

The inviscid assumption opens up several possibilities. A potential function, \( \Phi \), can be defined and Bernoulli's equation written along a streamline,

\[ v^2 \Phi = 0 \]  

(3.7.a)

\[ \frac{(v \Phi)^2}{2} + g(h-h_r) = \text{constant} \]  

(3.7.b)

The problem is that (3.7b) does not represent a boundary condition at the
diffuser since $h_r$ has no relationship to the water height at the
diffuser. The same problem is inherent with numerical solutions.
Hence a simplified one dimensional theory is attempted.

C. Jets Aligned with Current

A one dimensional theory is applied to the control sections of
Figure 3.2 in much the same manner as is done for propellers $(5,8)$.
Here a row of jets in Cartesian coordinates is substituted for a pro-
peller screw in cylindrical coordinates, and a new term $Q_o$ ($Q_o \ll Q_m$) is
included. Mixing flow is assumed to pass entirely between two dividing
streamlines forming the boundaries of the control sections. On the
boundaries and everywhere outside, pressure is assumed ambient. As
fluid moves through the diffuser its pressure is increased, but there is
no discontinuity in velocity. At section 1 the velocity is that of
undisturbed flow, $u_1 = u_r$ and $Q_1 = Q_m$. We would like to solve for $Q_m$.

When the momentum equation is applied between sections 1 and 4,
the only external force is the momentum from the diffuser, and assuming
$Q_o \ll Q_m$,

$$
\rho Q_1 u_1 + \rho Q_o u_o = \rho Q_1 u_4
eq (3.8)
$$

or

$$
Q_1 (u_4 - u_1) = Q_o u_o
$$

(3.9)

We would also like to write an energy equation, but not knowing the
amount of jet dissipation, energy can only be written between sections
1 and 2 and between sections 3 and 4,
Figure 3.2 Nozzles with Crossflow
\[ P_1 + \frac{1}{2} \mu_1 u_1^2 = P_2 + \frac{1}{2} \mu_1 u_2^2 \]  
(3.10)

\[ P_3 + \frac{1}{2} \mu_3 u_3^2 = P_4 + \frac{1}{2} \mu_4 u_4^2 \]  
(3.11)

Adding and setting \( P_1 = P_4 \) and \( u_2 = u_3 \),

\[ P_3 - P_2 = \frac{1}{2} \rho (u_4^2 - u_1^2) \]  
(3.12)

But \( P_3 - P_2 \) is also found by a force balance between sections 2 and 3,

\[ (P_3 - P_2)A = \rho Q_0 u_0 \]  
(3.13)

Combining (3.12) and (3.13) gives

\[ (u_4^2 - u_1^2) = \frac{2Q_0 u_0}{A} \]  
(3.14)

or

\[ u_4 = \sqrt{u_1^2 + \frac{2Q_0 u_0}{A}} \]  
(3.15)

Combining (3.9) and (3.14) permits

\[ Q_m = Q_1 = u_3 A = \frac{1}{2} A(u_1 + u_4) \]  
(3.16)

Thus the velocity through the diffuser, \( u_2 = u_3 \) is the average of the upstream and downstream velocities. Combining (3.15) and (3.16) and dividing by \( Q_o = n a_0 u_0 \) gives a formula for \( S^* \) (for \( Q_o \ll Q_m \)),

-35-
\[ S^* = \frac{Q_m}{Q_o} = \frac{1}{2} \frac{Q_c}{Q_o} + \frac{1}{2} \sqrt{\frac{Q_c^2}{Q_o^2} + \frac{2A}{n a_o}} \]  

(3.17)

The same \( \frac{Q_c}{Q_o} \) and \( \frac{A}{n a_o} \) that appear in (2.14) also appear in (3.17) but in different forms. In the limit of no crossflow, (3.17) becomes

\[ S^* = \sqrt{\frac{A/n}{2a_o}} \]  

(3.18)

and for large crossflows,

\[ S^* = \frac{Q_c}{Q_o} \]  

(3.19)

as expected.

I. Pressing from section 3 to 4, fluid undergoes a contraction. From (3.16) this is found to be

\[ c_c = \frac{u_3}{u_4} = \frac{A_4}{A} = \frac{1}{2} + \frac{1}{2} \frac{u_1}{u_4} \]  

(3.20)

which ranges from .5 at no crossflow to 1.0 at infinite crossflow. The width of the warm water zone from (3 ^a) and the velocity from (3.15) could be used as boundary conditions for describing the subsequent spreading beyond section 4. Unfortunately the longitudinal distance to section 4 is not known.
D. Jets Aligned Against the Crossflow

If the nozzles of Figure 3.2 are reversed to point into the current \((a=180^\circ)\), the situation is depicted in Figure 3.3. Relative to the current the diffuser momentum is now \(-\rho Q_o u_o\). Assuming no "expansion" losses as the flow goes from 1 to 4, the expression for dilution becomes

\[
S^* = \frac{Q_c}{Q_o} + \frac{1}{2} \sqrt{\left(\frac{Q_c}{Q_o}\right)^2 + \frac{2A/n}{a_o} \cos \alpha}
\]  
(3.21)

\[
S^* = \frac{Q_c}{Q_o} + \frac{1}{2} \sqrt{\left(\frac{Q_c}{Q_o}\right)^2 - \frac{2A/n}{a_o}}
\]

(3.22)

valid for \(Q_c > Q_{cr}\) defined by

\[
Q_{cr} = Q_o \sqrt{\frac{2A/n}{a_o}}
\]

(3.23)

For flow with \(Q_c < Q_{cr}\) crossflow is unable to penetrate the diffuser, (3.22) is invalid and the flow pattern looks like Figure 3.3d.

E. Jets Aligned at an Oblique Angle

Equation (3.21) can be modified to include small variations in \(\beta\) in Figure 3.1. As reasoned before, the crossflow which flows over the dormant diffuser is \(Q_c \cos \beta\). For small angles of \(\beta\) there is little deflection from the crossflow and the assumption of no forces along the dividing streamlines still holds. The generalized formula for dilution becomes,

\[
S^* = \frac{1}{2} \frac{Q_c}{Q_o} \cos \beta + \frac{1}{2} \sqrt{\left(\frac{Q_c}{Q_o}\right) \cos \beta^2 + \frac{2A/n}{a_o} \cos \alpha}
\]

(3.24)
Figure 3.3 Nozzles Against Crossflow
for small $\beta$. Just as in (3.22), a critical crossflow will occur for a
given combination of $\beta$ and $\alpha$. $(90^\circ \leq \alpha \leq 180^\circ)$. The critical crossflow is
given by

$$\left(\frac{Q_c}{Q_o} \cos \beta\right)^2 = \frac{-2A/n}{a_o} \cos \alpha \quad 90^\circ \leq \alpha \leq 180^\circ$$

(3.25)

At critical crossflow, crossflow and induced flow momentum are equal
and opposite and the discriminate will equal zero. For $Q_c > Q_{cr}$ crossflow
will penetrate the diffuser.

F. Alternating Nozzles

If the nozzles in Figure 3.1 alternate between $\alpha = 0^\circ$ and $180^\circ$,
there is no net momentum induced by the diffuser and $\alpha$ is effectively
equal to $90^\circ$. The volumetric dilution from (3.24) reduces to

$$S^* = \frac{Q_c}{Q_o} \cos \beta$$

The restriction $Q_c << Q_m$ is no longer necessary so accounting for the
condenser discharge,

$$S^* = -\frac{Q_c + Q_o}{Q_o} \cos \beta$$

(3.26)

A Discussion

Equation (3.24) predicts dilution as a function of a crossflow
term, $\frac{Q_c}{Q_o} \cos \beta$ and an induced flow term $\frac{2A/n}{a_o} \cos \alpha$. Figures 3.4 and 3.5
plot dilution and temperature rise against $Q_c/Q_o$ for $\beta = 0^o$ and $\alpha = 0^o$ and $180^o$ respectively. Typical prototype parameters are used:

$u_o = 10$ ft/sec, $a_o = 2$ ft$^2$, $Q_o = 20$ cfs, and $A/n = 600$ ft$^2$. 

-40-
Volumetric Dilution versus Crossflow

\[ u_0 = 10 \text{ ft/sec} \]
\[ a_o = 2 \text{ ft}^2 \]
\[ Q_o/n = 20 \text{ cfs} \]
\[ A/n = 600 \text{ ft}^2 \]
\[ \alpha = 0^\circ \]
\[ \beta = 0^\circ \]

Temperature Rise versus Crossflow

\[ u_0 = 10 \text{ ft/sec} \]
\[ a_o = 2 \text{ ft}^2 \]
\[ Q_o/n = 20 \text{ cfs} \]
\[ A/n = 600 \text{ ft}^2 \]
\[ \alpha = \beta = 0^\circ \]

Figure 3.4  Dilution and Temperature Rise versus Crossflow--Nozzles with Crossflow
Volumetric Dilution versus Crossflow

\[ u_0 = 10 \text{ ft/sec} \]
\[ a_0 = 2 \text{ ft}^2 \]
\[ Q_0/n = 20 \text{ cfs} \]
\[ A/n = 600 \text{ ft}^2 \]
\[ \alpha = 180^\circ \]
\[ \beta = 0^\circ \]

Dilution is Undefined

Temperature Rise versus Crossflow

\[ u_0 = 10 \text{ ft/sec} \]
\[ a_0 = 2 \text{ ft}^2 \]
\[ Q_0/n = 20 \text{ cfs} \]
\[ A/n = 600 \text{ ft}^2 \]
\[ \alpha = 180^\circ \]
\[ \beta = 0^\circ \]

\[ \frac{\Delta T_m}{\Delta T_o} \]

Figure 3.5 Dilution and Temperature Rise versus Crossflow—Nozzles against Crossflow
IV Experimental Program

Equation (3.24) predicts volumetric dilution $S^*$ in those cases where uniform conditions prevail downstream. It is therefore possible to predict downstream temperatures $T_m$ as a function of the entrained water temperature $T_a$ and the initial hot water temperature $T_o$ through equation (2.1). Stated another way, the volumetric dilution and temperature defined dilution (2.3) should be equal. Accordingly an experimental model was used to measure $T_m$ and $T_a$ for various geometries of crossflow and diffuser nozzles. The volumetric dilution predicted by (3.24) was then compared to the temperature defined dilution (2.3).

Five situations were modeled: 1) unidirectional nozzles pointed with the crossflow, 2) unidirectional nozzles pointed against the crossflow, 3) unidirectional nozzles at right angles to the crossflow, 4) alternating nozzles parallel to the crossflow, and 5) unidirectional nozzles subjected to an unsteady (tidal) crossflow. Most runs were simulations of the Shoreham prototype and are concentrated in the first and fourth classes. Furthermore, many runs with nozzles against the flow represent the realistic situation of insufficient crossflow to penetrate the diffuser line (see Figure 3.3d) and consequently uniform conditions were never obtained downstream. Only those runs creating uniform flow away of warm water are used in the analysis.

A. Modeling Considerations

Although these tests do not represent any particular prototype design, they were performed so that experimental results could be
scaled up to prototype dimensions without distortion. Three scaling laws should be observed.

1) The scaling should be undistorted, meaning vertical and horizontal length scales should be equal.

2) Densimetric Froude number should be equal in model and prototype.

3) Jet Reynold's numbers should be in the turbulent range for model and prototype.

Condition 1 is required of all near field temperature models (6) and requires that the scale ratio be large enough to model vertical dimensions in accordance with condition 3.

Condition 2 is required in all models where inertial and gravity (buoyancy) forces interact. $F_d$ is defined by equation (2.4) and for constant $\Delta T_o$ in model and prototype, model lengths, velocities, and times may be scaled according to

$$\frac{U_{o,m}}{U_{o,p}} = \left(\frac{m}{H}\right)^{\frac{1}{2}}$$

$$\frac{t_m}{t_p} = \left(\frac{m}{H}\right)^{\frac{1}{2}}$$

(4.1)

where $U_o$ is a characteristic velocity, $H_o$ is a characteristic length and $t$ is a characteristic time (appropriate with unsteady currents). $m$ and $p$ refer to model and prototype.

Condition 3 asks that model jets achieve full turbulence and is the most difficult condition to satisfy. A jet Reynolds number is
defined by

\[ R_j = \frac{u_o D_o}{v} \]  \hspace{1cm} (4.2)

where \( u_o \) is the nozzle discharge velocity, \( D_o \) is the nozzle diameter, and \( v \) is the kinematic viscosity of water. Pearce (6,4) summarizes the effects of Reynold's number on the characteristics of a circular jet, and concludes that for \( R_j \) greater than 3000, jets are fully turbulent over their full range and for \( R_j \) less than 3000, the jet remains laminar for an increasing distance along its centerline before turning turbulent. At \( R_j \) less than about 1000, there is virtually no turbulence and such a jet will not adequately model mixing from larger jets. Most of the experimental runs were performed at \( R_j \) about 1700, but turbulence was confirmed by injecting a small quantity of dye with the discharge and observing turbulent behavior.

In view of the modeling considerations above and the materials on hand, an undistorted Froude model with length scale 1:100 was chosen for the Shoreham model. Experimental results may be scaled up to Shoreham dimensions by multiplying all lengths by 100, velocities and times by 10 and discharges by 100,000.

**B. Physical Description of the Model**

The model is shown in Figure 4.1. Basin dimensions are 46'x28'x\( \frac{1}{2} \)'.

The hot water system consisted of tap water warmed by a steam heat exchanger and mixed in a constant head tank. Below the head tank two
parallel piping systems carried water through a pair of Brooks' rotameter flowmeters (capacity 6.3 GPM each) and into a pair of manifolds hidden under the false floor. Individual feeder lines carried water from the manifolds to interchangeable copper nozzles located on top of the floor. An equal flow of water was withdrawn from the basin sides to simulate plant intake. Temperatures were controlled by a temperature valve which controlled the quantity of tap water passing through the heat exchanger, and were monitored by thermistor probes inserted into the head tank and intake lines to the manifolds. Nozzle diameters were varied by interchanging nozzles, and to assist observation Rhodamine-B dye was injected from a constant head source into the manifold intake lines.

Crossflow was achieved by a circulating pumping system which included a 5HP, 250GPM centrifugal pump, 4" PVC piping, two Brooks' rotameters in parallel (capacity 66GPM each) and 20' by 4" diameter pipe manifolds extending across the upstream and downstream sections of the basin. Adjustable overtopping weirs and horsehair matting at both ends provided a uniform crossflow which was computed as the quotient of flowrate divided by channel crosssection area. For unsteady tests, the tide was broken into short intervals of constant velocity.

Temperatures were measured by an array of 90 Yellow Springs Instrument thermistor probes (time constant 7 secs, reproducibility 0.05°F.) which had been calibrated to compensate for small resistance differences corresponding to different length wire. Probes were located just below the surface and were scanned and printed on paper tape
by an automatic system designed by Digitec-United Systems.

C. Run Parameters

Table 4.1 lists run parameters and averaged temperatures, which will be explained later, for the steady runs.

Runs are broken into four blocks corresponding to nozzles aligned with a crossflow (runs 8-89), nozzles against the crossflow (runs 22-91), nozzles at right angles to the crossflow (runs 15 and 16), and alternating nozzles (runs 69-82).

D. Data Collection

Steady crossflow experiments were begun when a uniform crossflow of the desired velocity was achieved, the hot water system was primed with warm water of temperature about 20°F. above ambient, and an initial scan assured that all probes were at thermal equilibrium. To insure that water in the feeder lines was at the same temperature as water in the head tank, valves on the discharge lines were bled before starting. At the start of a run, discharge was adjusted to the correct condenser water flow, a small quantity of dye was injected, and a clock was started. Scans of the 90 probes took about three minutes each and were taken continuously shortly after the introduction of hot water. The experiment was continued until either the dye reached the downstream weir, or until the temperatures appeared to be at steady state, whichever came second. Usually runs were allowed to continue twenty to thirty minutes safely beyond the time needed for the dye to arrive downstream. Hence warm water from the discharge was recycled.
### Table 4.1 Experimental Run Parameters

**Block 1 Nozzles with Crossflow (α=β=0°)**

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**Block 2 Nozzles against Crossflow (α=180°; β=0°)**

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Block 3 Nozzles at Right Angles to Crossflow ($\alpha=0^\circ; \beta=90^\circ$)

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Block 4 Alternating Nozzles

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as crossflow, and later scans indicate slightly warmer ambient temperatures. This effect is accounted for in the data reduction.

E. Data Reduction

Data for each steady crossflow run consisted of an initial scan plus three to six subsequent temperature scans of each of the 90 probes. For each scan

1) calibration terms were added to the reading of each probe to produce accurate values of $T$ and $T_o$;

2) an ambient base temperature, $T_a$, was averaged from upstream probes behind the diffuser line, and

3) $\Delta T$ was calculated for each probe as $T - T_a$ and $\Delta T$ was calculated for the scan as $T_o - T_a$.

$\Delta T$'s for each probe were averaged over the scans and plotted. Figures 4.3 through 4.8 are examples. For unsteady runs, only one scan was taken for each time interval and the $\Delta T$'s for each interval were plotted. Examples can be found in reference (2).

The isothermal plots in Figures 4.3 through 4.8 are complex, especially in blocks 2 and 3 where the crossflow is oriented at right angles to or against the diffuser. Re-entrainment causes local temperature buildup which far exceeds downstream temperatures. Theory, however, only predicts well mixed downstream temperatures even for those cases where the immediate temperature rise may be of more significance. The following procedure was used to calculate $\Delta T_n$. In blocks 1 and 4 (nozzles aligned with the flow and alternating nozzles)
\( \Delta T_m \) was averaged from the \( \Delta T \)'s of those probes lying along a transverse line seven feet downstream from the diffuser. This was the first row of probes showing consistent uniformity. The averaging for Run 85 is shown in Figure 4.2. In block 2 (nozzles against the flow) only run 90 had \( Q_c < Q_{cr} \) and \( \Delta T_m \) was averaged along the same line seven feet from the diffuser. For block 3 \( \Delta T_m \) was averaged from probes along the dotted lines in Figures 4.6 and 4.7.

**F. Comparison of Theory with Experiment (Steady Crossflow)**

Figures 4.9 and 4.10 plot the experimental temperature rise against the theoretical dilution and temperature rise given by (3.24) for unidirectional nozzles (blocks 1,2,3). Figures 4.11 and 4.12 plot the experimental temperature rise against the theoretical dilution and temperature rise given by (3.26) for alternating nozzles.

For crossflows with the nozzles, data agrees well with (3.24). Only one run, 88, appears to deviate badly and the reason may be insufficient turbulence (low Reynolds number).

For crossflow against the nozzles, data is limited. A critical crossflow rate is observed, but there is insufficient data to determine if it is the value predicted by (3.23). (3.24) works well for the one run (90) with sufficient crossflow to penetrate the diffuser.

For crossflow at right angles to the diffuser \( (\theta = 90^\circ) \), (3.24) is less successful. Figures 4.6 and 4.7 show that increasing crossflow deflects the flow away at a steeper angle resulting in a narrower flow field with higher temperatures. Thus run 15 \( (u_r = .035 \text{ ft/sec}) \) agrees with the theory whereas run 16 \( (u_r = .070 \text{ ft/sec}) \) does not. Jet
well mixed temperature rise $\Delta T_m$
average along --- --- ---

$$\frac{1}{2} (0.8 + 0.6) = 0.7$$
$$0.7 = 0.7$$
$$0.7 = 0.7$$
$$\frac{1}{2} (0.4 + 1.2) = 0.8$$
$$0.7 = 0.7$$

$0.72$ ave.

Figure 4.2 Sample Calculation for $T_m$ (run 85)
Figure 4.3

RUN 85

$\alpha = 0^\circ$

$\beta = 0^\circ$

0.64 Ft/s

0.0
Figure 4.5

**Run 91**

\[ \alpha = \pi; \beta = 0 \]

\[ Q_c > Q_{cr} \]
Figure 4.9 Experimental Temperature Rise versus Theoretical Dilution (unidirectional nozzles)

\[
\frac{\Delta T_m}{\Delta T_0} = \frac{1}{2} \frac{a_e}{a_0} \cos \beta + \frac{1}{2} \sqrt{\left(\frac{a_e}{a_0} \cos \beta\right)^2 + \frac{2\alpha}{a_0} \cos\alpha}
\]
Figure 4.10
Experimental Temperature Rise vs Theoretical Temperature Rise
(Unidirectional Nozzles)

\[ \frac{\Delta T_m}{\Delta T_0} \]

\[ \frac{1}{2} Q_0 \cos \beta + \frac{1}{2} \sqrt{\left( \frac{Q_0}{Q_0} \cos \beta \right)^2 + \frac{2 A/n \cos \alpha}{Q_0}} \]
Figure 4.11
Experimental Temperature Rise versus Theoretical Dilution (Alternating Nozzles)
deflection violates the assumption of this theory (no force on the sides 
of the control volume) and hence (3.24) should be restricted to small 
values of \( \beta \). Since no experiments were performed for \( \beta \) between \( 0^\circ \) and 
\( 90^\circ \), a reasonable limit is hard to pick, but for \( 0^\circ \leq \beta \leq 45^\circ \), (3.24) 
should provide at least a qualitative picture. Additional problems 
with oblique orientations are the buildup of heat due to re-entrainment 
in a region near the diffuser and the difficulty of defining the 
downstream well mixed region. See Figures 4.6 and 4.7. Unless 
standards prescribe a mixing zone larger than the warm re-entrainment 
region, the most critical temperatures may occur before a well mixed 
temperature is achieved and hence would be unpredictable by the theory.

For crossflow parallel to alternating nozzles, data agrees well 
with (3.26). The most disagreement is found for low values of 
crossflow (runs 81, 19, 77, 79 have \( u_r \) less than or equal to \( 0.025 \text{ ft/sec} \)) 
where the prediction of (3.26) is conservative. This is to be expected 
since for low crossflow, diffusion of heat from the individual jets 
becomes significant relative to crossflow convection.

G. Application to Unsteady Crossflows

Temperatures calculated from (3.24) and (3.26) were compared 
with four tidal runs. Two runs were with alternating nozzles and two 
runs were with unidirectional nozzles. Because of size limitations, 
a complete tidal cycle could not be run. Instead two partial runs were 
performed for each configuration: a run from slack water through maximum 
flood (or ebb) and back to slack, and a run starting as ebb (or flood)
and reversing to flood (or ebb). The tidal cycle was divided into intervals during which velocities were held constant. Depths were never varied. The velocity versus time curves are shown in Figure 4.13.

Plots of isothermal rise during each interval can be found in reference (2). The hottest region for any run varies with time. Average temperatures, $\Delta T_m$ along the hottest transverse lines (parallel to the diffuser) are plotted in Figure 4.13 and are compared with the theoretical temperatures assuming that the system had been in continuous operation at the same crossflow velocity. The portions of the cycle where the diffuser is aimed against the crossflow are not plotted because of the inability of (3.24) to predict dilution in these cases.

Figure 4.13 indicates that except for some initial transience and a lag of about .1 hour (model time) prediction is quite good. Of course (3.26) is conservative for low crossflow velocities.
For small values of $\beta$ the following formula is predicted for dilution:

$$S^* = \frac{1}{2} \frac{Q_c}{Q_o} \cos \beta + \frac{\sqrt{\left(\frac{Q_c}{Q_o} \cos \beta \right)^2 + \frac{2A/n}{a_o} \cos \alpha}}{2}$$  \hspace{1cm} (3.24)$$

In this chapter the formula is applied to the design of a diffuser in a steady crossflow and in an unsteady crossflow.

**A. Steady Crossflow**

Assume $Q_o, u_r,$ and $h_r$ are given. To maximize dilution $\alpha$ and $\beta$ should equal zero, the nozzles should be aligned with the current, and (3.24) may be replaced by

$$S^* = \frac{1}{2} \frac{u_r h_r L}{Q_o} \sqrt{\left(\frac{u_r h_r L}{Q_o}\right)^2 + \frac{2bh_r u_r L}{Q_o}}$$  \hspace{1cm} (5.1)$$

Within constraints, dilution will increase with an increase in any of the independent variables, $b, u_o$ and $n$. $u_o$, however, is limited by the head on the condenser pump, $b/h_r$ should be limited to a reasonable ratio in order to achieve well mixed temperatures, and the cost of piping is a direct constraint on $bn$, the length of the diffuser.

**B. Unsteady (Tidal) Crossflow**

In the open sea, tidal velocities rotate, and in an open channel they are essentially harmonic and one dimensional. (3.24) can
be applied to certain portions of both cycles. In the open sea, assume
\( \alpha = 0^\circ \) and the angle \( \beta \) and the crossflow \( Q_c \) change periodically. Although
(3.24) is unverified for other than \( \beta = 0^\circ \), the following trends could
be predicted. Maximum dilution occurs with \( \beta = 0^\circ \) and the nozzles
oriented with the crossflow. As \( \beta \) increases, more and more
re-entrainment takes place and the downstream flow away zone becomes
narrower and warmer. For weak crossflows this trend should continue
for \( \beta \) up to \( 180^\circ \). For strong crossflows, a critical value of \( \beta \),
\( 90^\circ \leq \beta_{cr} \leq 180^\circ \) is defined

\[
\beta_{cr} = \cos^{-1} \left\{ \frac{2A/n \cos \alpha}{a_o} \left( \frac{Q_c}{Q_o} \right)^2 \right\} \quad 0^\circ \leq \alpha \leq 90^\circ 
\] (5.2)

At \( \beta_{cr} \) crossflow momentum and induced flow momentum are equal
producing very unstable flow with a maximum of re-entrainment. Beyond
\( \beta_{cr} \), a new regime is formed where crossflow penetrates the diffuser,
\( \alpha \) must be treated as \( 180^\circ \) and \( \beta \) must be measured from a new reference
\( 180^\circ \) away. The situation is easy to see if there is no net diffuser
momentum or \( \cos \alpha = 0^\circ \). \( \beta_{cr} = 90^\circ \) and dilution dies off from a maximum
at \( \beta = 0^\circ \) (crossflow perpendicular to the diffuser) to a minimum at
\( \beta = 90^\circ \) after which the pattern is symmetrical. Unfortunately no
model test were performed at widely varying \( \beta \).

Figure 4.13 shows the correlation between (3.24) and (3.26) and
experimental data for tidal variations in an open channel. Although
\( \beta \) is always zero, there is no analytic handle on the case where cross-
flow is against the diffuser. (\( \alpha = 180^\circ \)).
The problem associated with tidal flows periodically opposing the diffuser flow can be avoided by a safe design having alternating nozzles, thus guaranteeing a dilution of $Q_c(t)/Q_o$.

In order to predict temperatures using (2.1), the ambient temperature $T_a$ must be known. $T_a$ will be somewhat higher than $T_r$ in a tidal situation due to the return of heat after a tidal reversal. Because of unique boundaries at each site, $T_a$ can not be determined analytically, and a field study with dye is suggested. Reference (7) reports such a study for the Shoreham site. In this case the return of heat was small indicating that $T_a$ could be set equal to $T_r$ without losing much accuracy.

C. Conclusions

The shallow water diffuser is a special case in the study of thermal discharges. The assumptions of unstratified conditions and no friction lead to a simple formulation for volumetric dilution involving only lengths (areas), flow rates, and the number of nozzles; there is no dependence on Froude number. Crossflow and induced flow play important roles. Dilution is maximized when the two augment each other, while temperature buildup and unstable flow away persist when they are opposed. Equations (3.24) and (3.26) predict dilution for the case of unidirectional and alternating nozzles parallel to a current. (3.26) is conservative for a low crossflow. If a diffuser must be designed with nozzles pointing into a crossflow (as during a tidal cycle) experimental model studies may be needed to predict dilution.
REFERENCES


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Table

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</tr>
<tr>
<td>(Q_m)</td>
<td>mixing flow rate</td>
</tr>
<tr>
<td>(Q_r)</td>
<td>reference discharge rate</td>
</tr>
<tr>
<td>(Q_c)</td>
<td>crossflow rate</td>
</tr>
<tr>
<td>(Q_{cr})</td>
<td>critical crossflow rate</td>
</tr>
</tbody>
</table>
\( \dot{Q}_i \)  
induced flow rate

\( r, s \)  
cylindrical coordinates

\( R_J \)  
jet Reynolds's number

\( R_d \)  
diffuser Reynolds's number

\( S \)  
dilution

\( S^* \)  
volumetric dilution

\( t \)  
characteristic time

\( T \)  
local temperature

\( T_r \)  
reference temperature

\( T_a \)  
ambient temperature (of entrained water)

\( T_m \)  
downstream mixed temperature

\( T_o \)  
condenser water temperature

\( \Delta T_c(s) \)  
jet centerline temperature

\( \Delta T_o \)  
\( T_o - T_r \)

\( \Delta T \)  
\( T - T_r \)

\( \Delta T_m \)  
\( T_m - T_r \)

\( u, v, w \)  
Cartesian velocity components

\( u', v', w' \)  
dimensionless velocity components

\( u_r \)  
reference current velocity

\( u_o \)  
initial jet velocity

\( u_c(s) \)  
jet centerline velocity

\( U_o \)  
characteristic velocity

\( U_r \)  
characteristic flow away velocity

\( x, y, z \)  
Cartesian coordinates

\( x', y' \)  
dimensionless lengths

\( X \)  
basin length

\( Y \)  
basin width

\( \alpha \)  
nozzle inclination relative to horizontal

\( \beta \)  
crossflow approach angle

\( \beta_{cr} \)  
critical crossflow approach angle

\( \beta \)  
coefficient of thermal expansion

\( \varepsilon \)  
kinematic eddy viscosity
\( \rho \) density of water
\( \rho_a \) density of ambient water
\( \rho_o \) density of hot water
\( \Delta \rho_o \) \( \rho_a - \rho_o \)
\( \eta \) \( r/b(s) \)
\( \nu \) kinematic viscosity

**additional subscripts**

U,D refer to control sections (Figure 2.4)
1,2,3,4 refer to control sections (Figure 3.2,3.3)
m refers to model
p refers to prototype