CHAPTER 126

BUOYANT DISCHARGES FROM SUBMERGED MULTIPORT DIFFUSERS

by

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ABSTRACT

The application of submerged multiport diffusers for the discharge of
degradable liquid wastes and of heated cooling water from electric power
generation forms an important aspect of coastal zone management. Previous
buoyant jet models for submerged diffuser discharge have been developed for
the limiting case of discharge in unconfined deep water in the form of rising
buoyant jets. These models can be used for sewage diffusers, but are not
applicable for diffusers in shallow receiving water with low buoyancy, the
type used for thermal discharges ("thermal diffusers"). A multiport diffuser
will produce a general three-dimensional flow. Yet the predominantly two-
dimensional flow which exists in the center portion of the three-dimensional
diffuser can be analyzed as a two-dimensional "channel model". Theoretical
solutions for diffuser-induced dilutions are derived for the two-dimensional
case and verified experimentally. Furthermore, the theory can be applied to
the three-dimensional situation by requiring equivalency of far-field effects,
that is the frictional resistance governing the diffuser-induced motion at
larger distances from the diffuser line.

INTRODUCTION

In designing a system for the discharge of waste heat from electric energy
production the engineer has various alternatives at hand to control the thermal
effects within the receiving water. The choice of alternatives is influenced
by engineering, economic and environmental objectives, and may typically range
from low velocity surface discharges maximizing the heat transfer to the
atmosphere to high velocity submerged discharges maximizing the local mixing.
The increasing application of the latter type, in the form of a submerged multi-
port diffuser, stems primarily from the implementation of water quality standards
which require high dilutions within a limited mixing zone. The purpose of this
strategy of environmental conservation is to constrain the impact of heated
discharges to a small area.

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CHARACTERISTICS OF MULTIPORT DIFFUSERS

A submerged multiport diffuser is essentially a pipeline laid on the bottom
of the receiving water. The heated water is discharged in the form of round
turbulent jets through ports or nozzles which are spaced along the pipeline.
The jets actively entrain ambient water through shear effects at the jet
boundaries. An important feature is the interference of the individual round
jets of diameter D, spacing ℓ and velocity U_o a relatively short distance away
from the nozzles to form a two-dimensional jet, as shown in Figure 1. In usual
temperature prediction problems the major interest does not lie in the initial
region close to the nozzles, but rather farther away, for example at the surface
of the receiving water. It has been demonstrated (Reference 1) that outside
the initial region the jet dilution of the multiport diffuser is similar to an
"equivalent slot diffuser" with slot width

$$B = \frac{\pi D^2}{4\ell}$$

and equal discharge velocity U_o. Using the concept of the "equivalent slot
diffuser" reduces the number of dimensionless parameters characterizing a
multiport diffuser and thus provides a means to compare different diffuser
designs and applications.

For several decades coastal cities have utilized submerged multiport
diffusers for the discharge of municipal sewage water. Noteworthy aspects of
these "sewage diffusers" are: 1) Water quality standards dictate dilution
requirements in the order of 100 and higher when sewage water id discharged.
As a consequence these diffusers are limited to fairly deep water (more than
100 feet deep). 2) The buoyancy of the discharged water is significant. The
relative density difference between sewage water and ocean water is about 2.5%.

Only in very recent years have multiport diffusers been used for the
discharge of heated condenser cooling water from thermal power plants. Depend-
ing on the water quality classification of the receiving water and on the
cooling water temperature rise, dilutions between about 5 and 20 are required
within a specified mixing area. This dilution requirement frequently rules out
relatively simple disposal schemes, such as discharge by means of a surface
channel or a single submerged pipe. On the other hand, multiport diffusers
can be placed in relatively shallow water (considerably less than 100 feet deep)
and still attain the required dilutions. The economic advantage in keeping
the conveyance distance from the shoreline might be substantial, in particular
in lakes, estuaries or coastal waters with extended shallow nearshore zones.
"Thermal diffusers" have these characteristics: 1) They may be located in
relatively shallow water. 2) The buoyancy of the discharged water is low.
Initial density differences are in the order of 0.3% corresponding to a
temperature differential of about 20°F, an average value for thermal power
plants.

The difference in performance between "sewage" and "thermal" diffusers is
illustrated qualitatively in Figure 2. Figures 2a and 2c show the deep water
diffuser with high buoyancy which produces a distinct jet region with a stable
surface layer. Dilution prediction for this   situation   may be determined
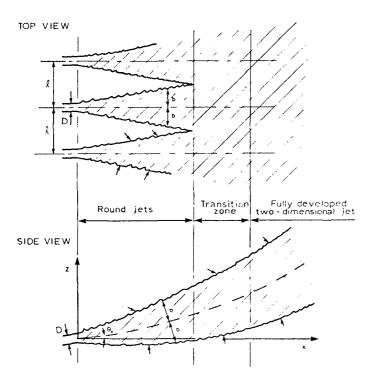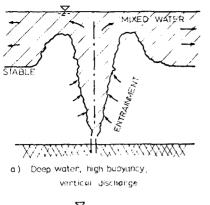
TOP VIEW

SIDE VIEW

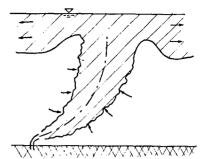Figure 1: Jet Interference for a Submerged Multiport Diffuser

using well-established buoyant jet models, such as Abraham (2), Fan and Brooks
(3), Hirst (4), in which the thickness of the surface layer is not a signifi-
cant portion of the total depth. On the other hand, diffusers in shallow water
with low buoyancy (Figures 2b and 2d) may not create a stable surface layer.
Subsequently, already mixed water is re-entrained into the jets decreasing the
dilution as would be predicted from buoyant jet models. As an extreme case of
boundary effects in shallow water a fully mixed flow zone may be established as
shown in Figure 2d.

     A theoretical and experimental investigation of the mechanics of multiport
diffusers in shallow water has been performed by Jirka and Harleman (5). The
study was concerned with a) The establishment of criteria which describe whether
stable or unstable near-field conditions will result, b) The development of a
predictive model for the case of an unstable near-field ("thermal diffuser")
and c) The investigation of three-dimensional circulations produced by a diffuser
of finite length in the receiving water which is either stagnant or flowing.
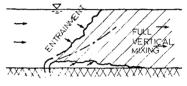
a) Deep water, high buoyancy,
      vertical discharge

c) Deep water, high buoyancy,
      non-vertical discharge

b) Shallow water, low buoyancy,
      vertical discharge

d) Shallow water, low buoyancy
      non-vertical discharge

Figure 2:   Qualitative Illustration of Vertical Flow Field in
            Diffuser Vicinity for Various Discharge Conditions

ANALYSIS

A multiport diffuser of length $2L_D$ will produce a three-dimensional flow field as depicted in Figure 3. Yet the predominantly two-dimensional flow which exists in the centerportion of the three-dimensional diffuser can be analyzed as a two-dimensional "channel model". The "channel model" consists of a diffuser section bounded by channel walls of length 2L and opening at both ends into a large reservoir (Figure 4).

The observed vertical structure of the flow field for a diffuser discharge within the two-dimensional channel is indicated in Figure 4 for the case of a stable near-field zone without re-entrainment. Four flow regions can be discerned in this general case:
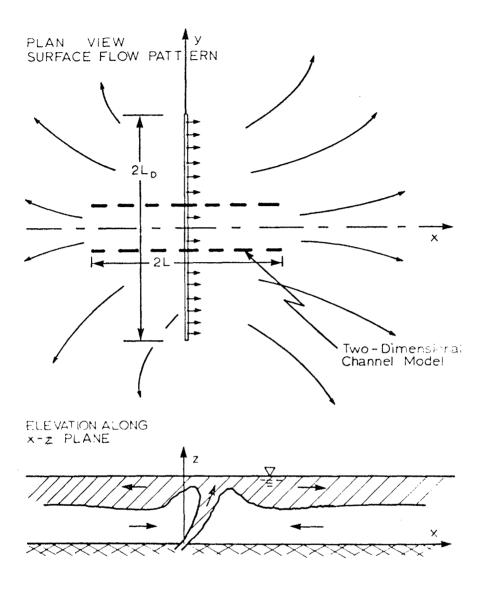
PLAN   VIEW
SURFACE FLOW PATTERN

y

$2L_D$

$2L$

x

Two - Dimensional
Channel Model

ELEVATION ALONG
x-z PLANE

z

x

Figure 3:   Three-Dimensional Flow Field for a Submerged
             Diffuser with Two-Dimensional Behaviour in
             Centerportion

1) Buoyant Jet Region: Forced by its initial momentum and under the action of gravity, the two-dimensional slot jet rises towards the surface entraining ambient water.

2) Surface Impingement Region: The presence of the free surface, with its density discontinuity, diverts the impinging jet in the horizontal directions.

3) Hydraulic Jump Region: An abrupt transition between the high velocity flow in the surface impingement region to lower velocities in the flow away zone is provided by an internal hydraulic jump.

4) Stratified Counterflow Region: A counterflow system is set up as a buoyancy-driven current in the upper layer and an entrainment-induced current in the lower layer.

Regions 1, 2 and 3 constitute the near-field zone; region 4 and the water body outside the channel, the far-field zone. The governing equations can be developed for each region accounting for its distinct hydrodynamic properties. Matching the solution for all regions provides an overall prediction of the diffuser induced flow field.

① Buoyant Jet Region
② Surface Impingement Region
③ Hydraulic Jump Region
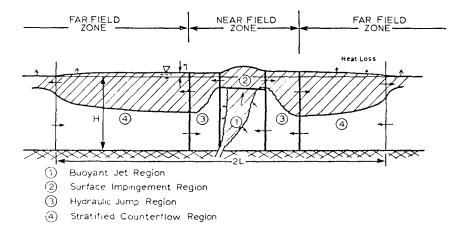④ Stratified Counterflow Region

Figure 4: General Vertical Structure of Diffuser Induced Flow Field (Two-Dimensional Channel Model)

Inspectional analysis of the governing equations shows that the following four dimensionless parameters characterize the multiport diffuser (equivalent slot B):

Densimetric Froude Number $F_s = \dfrac{U_o}{\sqrt{\dfrac{\Delta\rho_o}{\rho_a} gB}}$

Relative Submergence $\dfrac{H}{B}$

Angle of Discharge θ_o

Far-Field Parameter $\Phi = f_o \dfrac{L}{H}$

where

ρ_a = ambient density

$\Delta\rho_o$ = initial density difference

g = gravitational acceleration

H = water depth

f_o = bottom friction factor

Table 1 provides a survey of typical values of these parameters for both sewage and thermal diffusers.

THEORETICAL PREDICTIONS OF SURFACE DILUTIONS

Figure 5 gives theoretical prediction for surface dilutions as a function of F_s and H/B for the case of a vertical diffuser discharge ($\theta_o = 90^o$) with $\Phi = 1.0$. A criterion line divides the parameter range into two regions: diffusers with a stable near-field and diffusers with an unstable near-field. This is further illustrated in Figure 6. Diffusers with stable near-field have a distinct jet entrainment zone and the transition to the far-field is given by either a normal or a submerged internal hydraulic jump. Dilutions produced are essentially due to jet entrainment. In contrast, no internal hydraulic jump exists for diffusers with an unstable near-field. As a consequence, a local mixing zone with vertical recirculation is established and overall dilutions are governed by far-field effects: a dynamic equilibrium between buoyancy forces accelerating the flow away from the mixing zone and frictional forces retarding it.
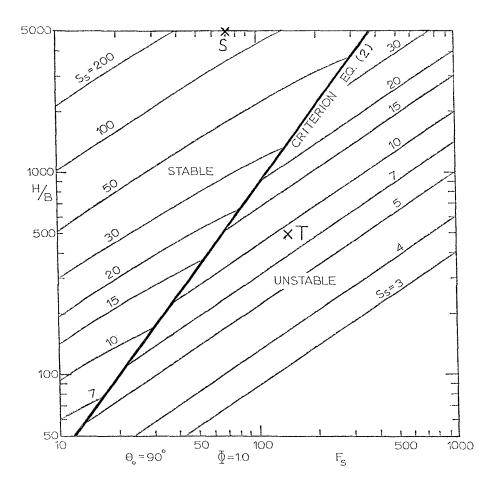
Figure 5: Surface Dilutions S_s as a Function of F_s, H/B Vertical Diffuser
(Typical values from Table 1 are indicated for a sewage diffuser
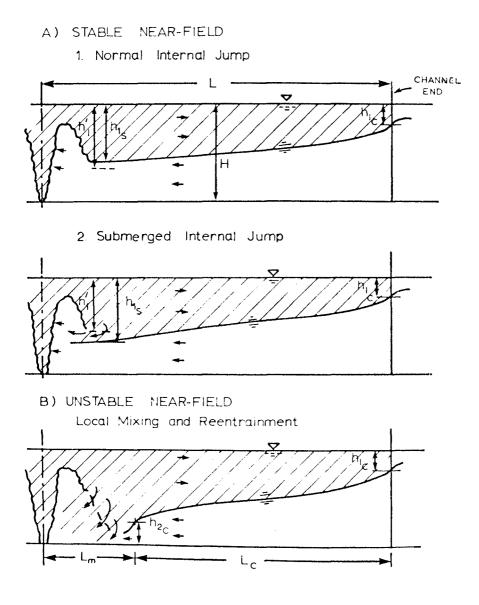S and a thermal diffuser T)

A) STABLE NEAR-FIELD

   1. Normal Internal Jump

   2. Submerged Internal Jump

B) UNSTABLE NEAR-FIELD

   Local Mixing and Reentrainment

Figure 6: Vertical Flow Conditions for Different Combi-
nations of Near-Field Stability and Far-
Field Effects

Table 1

Variables	Sewage Diffuser	Thermal Diffuser
Water depth, H (ft.)	100	20
Total discharge, Q_o (cfs)	400	1000
$\Delta\rho/\rho_a$	0.025	0.003
	(fresh-salt water)	($\Delta T_o \sim 20^\circ F$)
Total Diffuser Length,		
$2L_D$ (ft.)	3000	3000
Nozzle Diameter, D (ft.)	0.5	1.0
Nozzle Spacing, ℓ (ft.)	10	20
Discharge Velocity, U_o (fps)	6.8	8.5
Bottom Friction Coefficient, f_o	0.02	0.02
Equivalent Slot Width, B (ft.)	0.02	0.04
Dimensionless Parameters:		
F_s	70	140
H/B	5000	500
θ_o	variable	variable
Φ	0.1	1.5

The equation of the criterion line between stable and unstable conditions is

$$\frac{H}{B} = 1.84 \; F_s^{4/3} \tag{1}$$

The surface dilutions S_s in the stable parameter range are obtained from buoyant jet theory accounting for the thickness of the surface layer which is about 1/6 of the total depth. The surface dilutions S_s in the unstable parameter range are determined by stratified flow theory (far-field effects) and can be written as

$$S_s = 1.6 \; (\frac{k}{F_s})^{2/3} \; \frac{H}{B} \tag{2}$$

where k is a function of the far-field parameter Φ and may be tabulated over the range of interest.

Table 2

Φ	0	0.1	0.5	1.0
k =	0.25	0.225	0.187	0.17

The parameter values for a typical sewage diffuser (5) and a typical thermal diffuser (T) from Table 1 are indicated in Figure 5. The different hydrodynamic conditions and resultant surface dilution for the two diffuser types are clearly evident. For the thermal diffuser a dilution of 9 is predicted.

The dilution predictions of Figure 5 can be compared to Figure 7, in which buoyant jet theory (accounting for surface layer thickness) has been used over the whole parameter range. A dilution of 14 would be predicted for the thermal diffuser. This considerably higher prediction is due to the fact that the presence of the local mixing zone and re-entrainment has not been considered and would lead to drastically erroneous designs.

The results of the vertical discharge ($\theta_o = 90^o$) are also applicable to diffusers with alternating nozzles which do not have any net horizontal momentum and produce a similar symmetric flow field. Analysis and results for non-vertical discharges are also given in Reference 5. The theoretical results have been verified in an extensive series of experiments.

THREE-DIMENSIONAL ASPECTS OF THERMAL DIFFUSERS

The deep water, high buoyancy type diffuser with stable near-field always produces a well-stratified flow away from the diffuser line in all directions (see Figure 2). A contrasting behaviour can be observed in thermal diffusers in shallow water with low buoyancy as shown in Figure 8 for a vertical discharge: The presence of the entrainment mechanisms in combination with the near-field instability gives rise to a strong current which sweeps along the diffuser axis toward the center of the diffuser and then departs perpendicular to the diffuser line. The presence of this circulation leads to undesirable repeated re-entrainment and reduced dilutions. The circulation can be prevented by providing some discharge momentum along the diffuser axis through use of alternating nozzle with variable horizontal orientation β. In particular the orientation

$$\beta(y) = \cot^{-1}[\frac{1}{\pi} \log \frac{(1 + \frac{y}{L_D})}{(1 - \frac{y}{L_D})}] \qquad (3)$$

where y = distance along diffuser axis (Figure 8) was derived to provide stratified conditions with flow-away in the upper layer in all directions and maximum dilution.

Figure 9 shows experimental results from a three-dimensional diffuser located in a laboratory basin. The lower basin boundary is the symmetry line perpendicular to the diffuser axis: a mirror image can be assumed below this line. Horizontal dimensions x, y are normalized by the water depth H. Figures
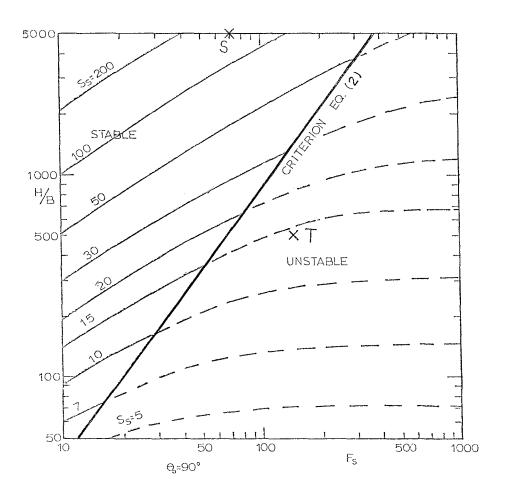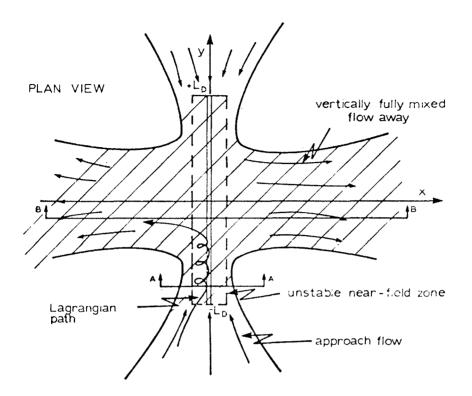
Figure 7: Theoretical Surface Dilutions S_s Assuming Buoyant Jet Theory Over the Whole Parameter Range (typical values from Table 1 are indicated for a sewage diffuser S and a thermal diffuser T)
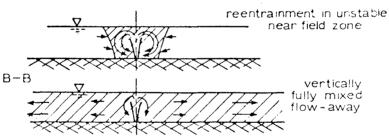
PLAN VIEW

vertically fully mixed
flow away

unstable near-field zone

Lagrangian
path

approach flow

A - A

reentrainment in unstable
near field zone

B - B

vertically
fully mixed
flow - away

Figure 8:   Three-Dimensional Flow Field for a Multi-
            port Diffuser in Shallow Water with Low
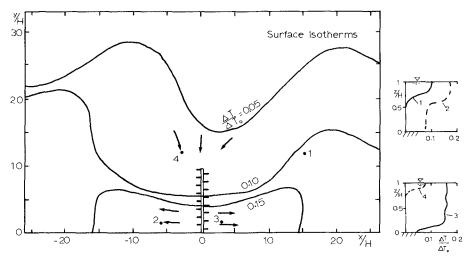   Buoyancy and Vertical Discharge (No Control)

Figure 9a: Nozzle Orientation Normal to Diffuser Axis, F_s = 158, H/B = 628, Φ = 0.9
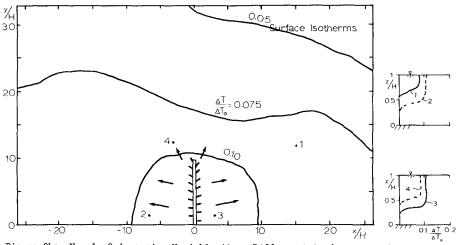

Figure 9b: Nozzle Orientation Variable Along Diffuser Axis (Equation 3), F_s = 160, H/B = 628, Φ = 0.9

Figure 9: Experimental Results for a Multiport Diffuser with Alternating
           Nozzles in Stagnant Water. Shows Effect of Horizontal Nozzle
           Orientation on Horizontal and Vertical Temperature Distribu-
           tions. Arrows Indicate Surface Currents.

9a and 9b show the normalized surface isotherms $\frac{\Delta T}{\Delta T_o}$, where ΔT_o = initial temperature difference, ΔT = temperature difference at the surface, and observed surface flow patterns for different horizontal nozzle orientations, but otherwise identical discharge conditions (same F_s, H/B and Φ). In Figure 9a the diffuser has alternating nozzles all normal to the axis thus providing no control and giving rise to a horizontal circulation. In Figure 9b the alternating diffuser nozzles are oriented according to Equation (3) thus pre-venting the horizontal circulations. The difference in stratification can be seen from the vertical temperature profiles at four points in the diffuser vicinity. The diffuser with control guarantees considerably better dilutions.

Under conditions of a controlled three-dimensional flow field the dilution prediction of the two-dimensional "channel model" can also be applied to the three-dimensional diffuser case provided that $L \approx L_D$ in the definition of the far field parameter Φ. In other words, the dilution in the two-dimensional channel will be equivalent to the three-dimensional diffuser if the channel length is taken about equal to the diffuser length.

EFFECT OF CROSSFLOW

If a multiport diffuser is placed in a steady cross current with magnitude u_a (see Figure 10) then the diffuser-induced flow field is modified by the crossflow. Two additional dimensionless parameters are needed to characterize the problem:

Volume flux ratio $V = \dfrac{u_a H}{U_o B}$

Angle of diffuser
with direction of γ
crossflow

Extremal cases of crossflow are $V = 0$ (stagnant conditions, as treated in the previous paragraphs) and V = large, such as in river applications, which result in full mixing, so that the dilution $S_s = V$. The diffuser was studied under moderate crossflow conditions which are important in lakes or coastal applica-tions. A strong dependence on diffuser angle γ was found: Diffusers parallel to the crossflow ($\gamma = 0°$) in general produce lower dilutions as compared to diffusers perpendicular to the crossflow ($\gamma = 90°$). Figure 11 shows an example for a diffuser with alternating nozzles and nozzle orientation given by Equation (3): the extent of the $\Delta T/\Delta T_o = 0.075$ isotherm is considerably smaller for the perpendicular diffuser (Figure 11a).

DESIGN CONSIDERATIONS

Practical thermal diffuser design involves the geometric outlay and dimensioning of a diffuser for a discharge flow Q_o, a temperature rise T_o and subject to an allowable surface temperature rise at the edge of some mixing zone, $T_{s\ max}$. The required minimum surface dilution is then

$$S_{s\,min} = \frac{\Delta T_o}{\Delta T_{s\,max}} \tag{4}$$

The designing engineer usually has to choose between a diffuser with alternating nozzles (zero net horizontal momentum) and a diffuser with unidirectional nozzles (net horizontal momentum). This choice is dependent on the ambient current system and bathymetry. From experimental observations it appears that, unless the ambient currents are steady and strong and/or the bathymetry has a significant offshore slope, unidirectional diffusers always tend to produce circulations in the diffuser area which cause undesirable re-entrainment of already heated water. Alternating diffusers on the other hand tend to produce a stratified flow (except within the unstable near-field zone) with a reduced tendency for re-entrainment. As most sites are characterized by unsteady, possibly reversing currents (tidal or wind-driven) and by extended shallow near-shore flats, the installation of an alternating diffuser system seems to have certain advantages.

As it is recognized that thermal diffusers will as a consequence of their dynamic characteristics always produce an unstable near-field, the following design considerations for alternating diffusers can be given:

a) The temperature field should be uniform along the diffuser line. Theory indicates that in case of variable depth the discharge per unit length should be varied proportional to the 3/2 power of depth to produce uniform surface dilutions.

b) Currents should be prevented from sweeping along the diffuser line. In case of weak or no currents the nozzle orientation should be varied along the line (Equation (3)). In case of stronger currents alignment of the diffuser axis parallel to the current direction should be avoided.
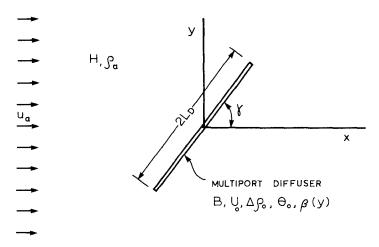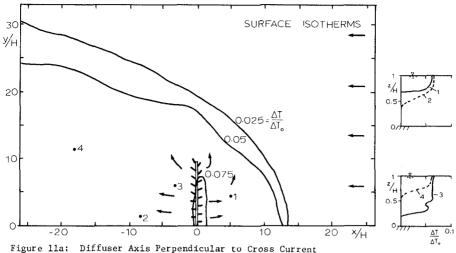

Figure 10: Multiport Diffuser in Ambient Crossflow

Figure 11a: Diffuser Axis Perpendicular to Cross Current
F_s = 69, H/B = 558, Φ = 0.4, $\beta(y)$ Eq. (3), V = 14.0, γ = 90°

Figure 11b: Diffuser Axis Parallel to Cross-Current
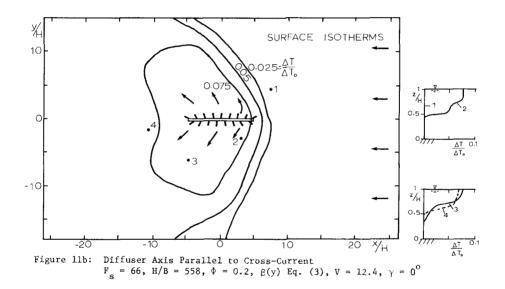F_s = 66, H/B = 558, Φ = 0.2, $\beta(y)$ Eq. (3), V = 12.4, γ = 0°

Figure 11:   Experimental Results for a Multiport Diffuser with Alternating
Nozzles in a Cross-Current. Shows Effect of Diffuser Alignment.

c) In case of variable ambient conditions (such as unsteady wind-driven or tidal currents) it is desirable to use a diffuser which promotes stratification and is effective for both current directions. This objective is met by using diffusers with alternating nozzles (no net horizontal momentum). For these diffusers the required total length, $2L_D$, can be estimated by virtue of Equation (2) as

$$2L_D = \frac{Q_o}{(\frac{\Delta\rho_o}{\rho_a} g H_{ave}^3)^{1/2}} \frac{S_{s\ min}^{3/2}}{(4k)^{1/2}} \tag{5}$$

where H_{ave} = average depth in the discharge area and k is given in Table 2 as a function of $\Phi = f_o L_D/H$.

If the plant characteristics, Q_o and ΔT_o, and the water depth, H_{ave}, are taken as fixed, then Equation (5) states the attained dilution, $S_{s\ min}$, is uniquely dependent on the length, $2L_D$. Considering Figure 7, this places the design along an   isoline S_s = const, for the example of Table 1, S_s = 9. The final position on the isoline is dependent on the choice of the secondary diffuser characteristics, namely discharge velocity, U_o and dimension, B. It is desirable to place the design close to the stable region in order to promote stratification by minimizing the intensity of the vertical recirculating eddy in the near-field. Shifting the diffuser design close to the stable region, however, necessitates lowering of the discharge velocity, U_o. A practical lower limit on U_o is about 5 fps due to the fact that lower discharge velocities require large diffuser pipes in order to maintain uniformity of discharge along the diffuser line. Thus from the point of view of discharge velocity, the thermal diffuser (T) of Figure 7 and Table 1 is about optimal (U_o = 8.5 fps). Another interesting consequence of Equation (5) is the fact that the diffuser design depends only on the total waste heat rejection, H_R, of the power plant (dependent on plant efficiency) and not on the particular condenser design (i.e. choice of Q_o and ΔT_o). Using the expression for the waste heat rejection

$$H_R = \rho c_p Q_o \Delta T_o$$

and a linear density-temperature relationship

$$\frac{\Delta\rho_o}{\rho_a} = |\beta\Delta T_o|$$

where β is the coefficient of thermal expansion, Equation (5) can be re-written in the form

$$2L_D = \frac{H_R/\rho c_p}{(\beta g H_{ave}^3)^{1/2} (4k)^{1/2}} \frac{1}{(\Delta T_{max})^{3/2}} \tag{6}$$

In summary, Equation (6) indicates that the required diffuser length of an alternating diffuser is only a function of the waste heat rejection, H_R, the available water depth, H_{ave}, the far-field condition, k, and the imposed temperature standard, ΔT_{max}.

With proper schematization of the site geometry, the theoretical predictions of Reference 5 can be used to provide a diffuser design or preliminary design estimate for the screening of alternative discharge schemes and/or for further investigation in a hydraulic scale model.

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