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BUOYANT JETS IN CONFINED SURROUNDINGS

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Abstract

Jet analysis by means of the ordinary differential equations, obtained through integration of the governing equations normal to the jet trajectory, is an efficient and accurate technique for the evaluation of thermal discharges. However, in certain instances some of the underlying assumptions, such as small spreading and negligible pressure gradients, may be violated and hence the applicability of the analysis restricted. a) In the heated surface jet, buoyancy exerts a distorting influence on jet behaviour, promoting lateral spreading. It is shown that available techniques are only capable to predict the transition from non-buoyant to more buoyant behaviour. The limiting case of buoyancy dominated flow cannot be predicted reliably. b) Submerged buoyant jets in water of finite depth may exhibit vertical instabilities following impingement upon the free surface. In case of instability, recirculation of already mixed water into the jet zone will occur. Criteria which determine the stable and unstable range as a function of governing parameters are presented. Thermal multiport diffusers are usually characterized by an unstable, recirculating near-field and hence simple buoyant jet analyses are not valid.

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DISCLAIMER

Introduction

Temperature prediction models involve the determination of the spatial and temporal distribution of the discharge into a water body of waste heat from electric power generation. This distribution is governed by different heat transport mechanisms which are a function of the discharge characteristics and of the hydrologic conditions prevailing in the disposal area.

The problem of temperature prediction for waste heat disposal is distinguished by the simultaneous occurrence of two factors which add a degree of complexity beyond that for other predictive models for effluent dispersion and water quality in lakes, rivers or coastal waters or air quality in the atmosphere. These two factors are a) the buoyancy of the discharge and b) the considerable volume and momentum of the effluent. The buoyancy of the discharge which is associated with the temperature change as the cooling water passes through the power plant condenser requires the simultaneous determination of both fluid motion (velocity distribution) and heat distribution within the water body. The volume and momentum of the discharge will in general affect the ambient flow field. Hence, it is not possible to consider the effluent as a passive tracer introduced into the ambient flow. This is particularly the case in a zone close to the discharge area (near-field zone) where advection and free turbulence created by the shearing action of the discharge with respect to the ambient water causes jet diffusion of the heated water. Outside this immediate near-field exists a considerably larger far-field zone in which the heat is distributed by buoyancy driven currents and through diffusion and advection by ambient currents.

Predictive models for hydrothermal analysis can broadly be classified into two groups: complete models and zone models. In the complete models the governing equations are solved in their more general form over the whole region of interest. These models promise a significant advance through the use of modern computers with high speed and large memories. Yet several problems have to be recognized: 1) The state of the art in computational techniques requires many simplifying assumptions regarding turbulent fluid flow and heat fluxes. 2) The flow and temperature field induced by a thermal discharge exhibits distinctly different hydrodynamic zones. Consequently, the simplifying assumptions utilized in the formulation of the complete model are not uniformly valid throughout the region of interest. This may cause considerable error and thus restrict the utility of such computer models. 3) Boundary conditions at the edge of the solution domain, notably open fluid boundaries, are difficult to specify.

In the zone models the whole region of interest is divided into several zones with distinct hydrodynamic properties (such as near-field and far-field). For each zone it is then possible to simplify the governing equations by dropping unimportant terms (through a formal scaling process). This gives a considerable advantage in the mathematical treatment and improved accuracy in the solution. Despite this advantage, problems remain inasmuch as utmost care has to be taken whether some of the assumptions which yield simplified governing equations are not violated in the actual application. Furthermore, there may be a lack of criteria on how to establish a correct division of the whole region into zones.

In this paper jet diffusion models are discussed as one class of zone models which are of particular importance in the prediction of the near-field behaviour of heated discharges. The simplifying assumptions pertinent to jet diffusion are discussed. Subsequently, the restrictiveness of these assumptions in the development of actual buoyant discharge models is analyzed. This is done for two types of models: buoyant surface discharges in deep receiving water and buoyant submerged discharges in shallow receiving water.

Jet Diffusion

A special class of fluid motion associated with shear-generated free turbulence is commonly referred to as jet diffusion. Dominating transport processes in jets are the convection by the mean velocities along the trajectory of the jet and the lateral turbulent diffusion normal to the jet trajectory through the irregular eddy motion within the jet. The convective mechanism is due to the initial discharge momentum and/or the vertical acceleration in the case of submerged buoyant jets.

Main properties of the jet flow field are deduced from experimental observations:

1) Gradual spreading of the jet along the trajectory. The jet width is small compared to the distance from the discharge. This allows the typical boundary layer type approximations.

2) Self-similarity of the flow. Except for an initial transition zone, the transverse profiles of velocity and temperature at different axial locations are similar to each other. Typically a bell-shaped distribution is found. Local jet quantities can be expressed as a function of centerline quantities and jet width.

3) Fluctuating turbulent quantities are small compared to mean flow quantities.

4) Pressure gradients, both transverse and longitudinal, are small. This is true only for jets issuing into relatively large unconfined regions.

In general, the governing boundary layer type equations are formulated in a local coordinate system following the trajectory of the jet. Exact similarity solutions to these equations can be found if semi-empirical mixing length assumptions are made (Schlichting (1)). For engineering purposes, however, it is more practical to specify similarity profiles a priori. By integrating across the jet the governing partial differential equations are then easily reduced to ordinary ones with the axial distance as the independent variable. This integral technique (method of moments) has been found useful and sufficiently accurate in many applications. Examples include buoyant jets in deep (unconfined) receiving water, either non-stratified or stratified. A further advantage of the integral technique is the possibility to consider a flowing receiving water by defining a gross force acting on the jet. As in all problems of turbulent flow, empirical coefficients appear in the analysis and have to be determined from experiments.

It must be expected that the jet diffusion analysis will break down as a viable predictive tool whenever one of the underlying assumptions, 1) to 4), is severely violated.

Buoyant Surface Jets

Three-dimensional predictive models of buoyant surface jets which take account of the underlying transport phenomena have been proposed by Stolzenbach and Harleman (2,3), Prych (4) and Stefan and Vaidyaraman (5). The theoretical premises on which these models are built are examined herein.

Governing equations: Figure 1 defines the problem under consideration: Discharge parallel to the free surface of the receiving water which is deep and quiescent.

Following Stolzenbach and Harleman the governing steady-state equation can be written as:

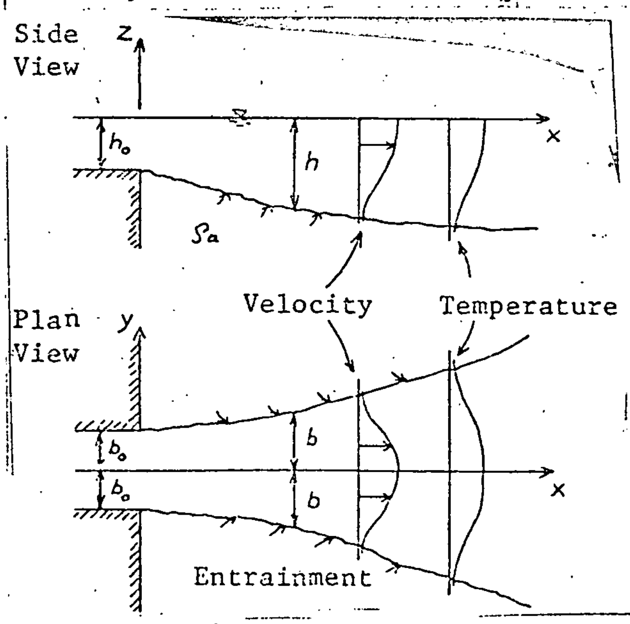
$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{1}$$

$$\frac{\partial u^2}{\partial x} + \frac{\partial uv}{\partial y} + \frac{\partial uw}{\partial z} = + \frac{g}{\rho_a}\int_z^{-\infty} \frac{\partial \Delta\rho}{\partial x}\,dz - \frac{1}{\rho_a}\frac{\partial p_d}{\partial x} - \frac{\partial \overline{u'^2}}{\partial x} - \frac{\partial \overline{u'v'}}{\partial y} - \frac{\partial \overline{u'w'}}{\partial z} \tag{2}$$

$$\frac{\partial uv}{\partial x} + \frac{\partial v^2}{\partial y} + \frac{\partial vw}{\partial z} = + \frac{g}{\rho_a}\int_z^{-\infty} \frac{\partial \Delta\rho}{\partial y}\,dz - \frac{1}{\rho_a}\frac{\partial p_d}{\partial y} - \frac{\partial \overline{u'v'}}{\partial x} - \frac{\partial \overline{v'^2}}{\partial y} - \frac{\partial \overline{v'w'}}{\partial z} \tag{3}$$

$$\frac{\partial uw}{\partial x} + \frac{\partial vw}{\partial y} + \frac{\partial w^2}{\partial z} = \frac{1}{\rho_a}\frac{\partial p_d}{\partial z} - \frac{\partial \overline{u'w'}}{\partial x} - \frac{\partial \overline{v'w'}}{\partial y} - \frac{\partial \overline{w'^2}}{\partial z} \tag{4}$$

$$\frac{\partial uT}{\partial x} + \frac{\partial vT}{\partial y} + \frac{\partial wT}{\partial z} = - \frac{\partial \overline{u'T'}}{\partial x} - \frac{\partial \overline{v'T'}}{\partial y} - \frac{\partial \overline{w'T'}}{\partial z} \tag{5}$$



$$\Delta\rho = \beta(T_a - T) \tag{6}$$

Fig.1: Buoyant Surface Jet

where x, y, z = Cartesian coordinates u, v, w = time-averaged velocities in x, y, z; u', v', w' = turbulent velocity fluctuations; T = time-averaged temperature; T' = turbulent temperature fluctuation; T_a = ambient temperature; $\Delta\rho$ = local density difference; ρ_a = ambient density; g = gravitational acceleration; β = coefficient of thermal expansion. The pressure components appearing in the governing equations are of interest: 1) The dynamic pressure, p_d, results from the balance with the vertical inertial and diffusive terms in Eq. 4. 2) The pressure term, $g \int_z^{-\infty} \Delta\rho dz$, arises from the buoyancy of the flow combined with the fact that the free surface prevents any upward motion of the flow, unlike submerged buoyant jets. This pressure force is the primary mechanism which significantly affects the behaviour of surface buoyant jets as compared to non-buoyant jets. An additional difference is due to the buoyant damping of vertical turbulent fluctuations, therefore reducing vertical spreading and entrainment.

Scaling: By inspection of the equations, the obvious scaling parameter, which determines the importance of the buoyancy term, is derived as a local densimetric Froude number,

$$F_L = u_c \left(\frac{\Delta\rho_c}{\rho_a} gh\right)^{-1/2} \qquad (7)$$

where u_c, $\Delta\rho_c$ are values of u and $\Delta\rho$ at the jet axis on the surface and h is some measure of the jet depth. A characteristic feature of buoyant jets is the monotonic decrease of F_L along the axis. The most elementary indication of this feature is obtained by using the axisymmetric non-buoyant jet solutions for the limiting case of a slightly buoyant (F_L large) jet: $u_c \sim x^{-1}$, $h \sim x$, $\Delta\rho_c \sim x^{-1}$, therefore $F_L \sim x^{-1}$. Hence, to describe buoyant jets the dominant terms over the whole range of F_L, from $F_L \rightarrow \infty$ to $F_L \rightarrow 1$, have to be retained.

The dominant terms for these limiting cases are derived through scaling. For this purpose, s, b, h and u^*, v^*, w^* are taken as the characteristic lengths and velocities, respectively, in the x, y, z directions. $\Delta\rho^*$ is the characteristic density difference. Furthermore, throughout the scaling process the following assumptions are made: i) the scale of turbulent covariances is small compared to the mean convection, e.g., $\overline{u'u'}/u^{*2} \sim \delta$, which is consistent with all turbulent shear flows, and ii) the dynamic pressure scale is $p_d^* \sim \rho_a w^{*2}$.

$F_L \to \infty$: In this case, the buoyant term is negligible, and the typical jet diffusion pattern will result, characterized by $b/s \sim \delta$ and $h/s \sim \delta$ and, as a consequence, by $v^*/u^* \sim \delta$ and $w^*/u^* \sim \delta$. The governing equations are Eqs. 1, 6 and

$$\frac{\partial u^2}{\partial x} + \frac{\partial uv}{\partial y} + \frac{\partial uw}{\partial z} = - \frac{\partial \overline{u'v'}}{\partial y} - \frac{\partial \overline{u'w'}}{\partial z} \tag{8}$$

$$\frac{\partial uT}{\partial x} + \frac{\partial vT}{\partial y} + \frac{\partial wT}{\partial z} = - \frac{\partial \overline{v'T'}}{\partial y} - \frac{\partial \overline{w'T'}}{\partial z} \tag{9}$$

These classical isothermal jet equations express simply the balance between convective transport of axial velocity, u, or temperature, T, and transverse and vertical diffusion. The temperature acts here like a tracer substance, without dynamic effect. The other momentum Eqs. 3 and 4, reduce to a trivial balance between dynamic pressure and diffusion, and have not been written.

$F_L \to 1$: In this case, the buoyant term is of the same order of magnitude as the convective terms. As the buoyant pressure gradient acts in both horizontal directions, the flow field will behave as $b/s \sim 1$ and thus $v^*/u^* \sim 1$. A problem remains regarding the relative magnitude of h/s. As an initial assumption, one might take $h/s \sim \delta$, i.e., the vertical extent of the flow field is only gradually varying, hence $w^*/u^* \sim \delta$. The validity of this assumption is discussed later.

$$\frac{\partial u^2}{\partial x} + \frac{\partial uv}{\partial y} + \frac{\partial uw}{\partial z} = + \frac{g}{\rho_a} \int_z^{-\infty} \frac{\partial \Delta\rho}{\partial x}\, dz - \frac{\partial \overline{u'w'}}{\partial z} \tag{10}$$

$$\frac{\partial uv}{\partial x} + \frac{\partial v^2}{\partial y} + \frac{\partial vw}{\partial z} = + \frac{g}{\rho_a} \int_z^{-\infty} \frac{\partial \Delta\rho}{\partial y}\, dz - \frac{\partial \overline{v'w'}}{\partial z} \tag{11}$$

$$\frac{\partial uT}{\partial x} + \frac{\partial vT}{\partial y} + \frac{\partial wT}{\partial z} = - \frac{\partial \overline{w'T'}}{\partial z} \quad (12)$$

The vertical momentum equation, 4, can be neglected. In contrast to the equation for $F_L \to \infty$, only the vertical diffusion terms are significant for $F_L \approx 1$. Thus, in this limiting case, the transverse flow velocity and temperature distributions will not have a jet-like sheared profile. Furthermore, the shear term, $\partial\overline{v'w'}/\partial z$, which acts on the lateral flow emerges as a dominant term.

Integral analysis: Profile assumptions on the lateral and vertical distribution of u, v and T (or $\Delta\rho$) have to be made so that the preceding partial differential equations can be reduced to ordinary differential equations (jet integral analysis). However, as the equations should describe the typical transition in surface buoyant jets from large to small values of F_L, the following difficulties will arise: a) Only the vertical jet profiles are truly of shear type. The transverse profiles change from shear type to a more uniform distribution (shear acting only at the edges). b) The distribution of the lateral velocity, v, is not readily specified in terms of centerline quantities. Some hypothetical assumptions have to be made in this respect.

Despite these theoretical restrictions it may be useful to retain the assumptions of jet-like lateral profiles in order to describe the deviation due to buoyancy from the more non-buoyant behaviour, without attempting to describe the limiting case of strongly buoyant behaviour. The utility and range of applicability of such an approach have to be demonstrated mainly by comparison to experiments. If this procedure is followed and the distributions for velocity and density

$$u/u_c = f(\eta,\zeta), \; \Delta T/\Delta T_c = \Delta\rho/\Delta\rho_c = g(\eta,\zeta),$$

where $\eta = y/b$ and $\zeta = z/h$ and f and g define bell-shaped jet distributions, are assumed, then after transverse integration the governing equations acquire the following general form

$$\frac{dQ}{dx} = c_1\alpha_o u_c h + c_2\alpha_v u_c b \quad (13)$$

$$\frac{dM}{dx} = \frac{dP}{dx} \tag{14}$$

$$\frac{dH}{dx} = -c_3 k \Delta T_c b \tag{15}$$

$$\frac{db}{dx} = \varepsilon + \left(\frac{db}{dx}\right)_B \tag{16}$$

where c_1, c_2 and c_3 are profile-dependent coefficients, k is a coefficient for surface heat loss and ε is the rate of spreading for a non-buoyant jet.

The above equations are well amenable to numerical solution using appropriate initial conditions at the discharge point. The integral quantities are defined as follows:

Volume flux $$Q = \int_A u \, d\eta \, d\zeta \tag{17}$$

Momentum flux $$M = \int_A \rho_a u^2 \, d\eta \, d\zeta \tag{18}$$

Pressure force $$P = \int_A \left[\int_{-\infty}^{\zeta} g\Delta\rho \, d\zeta\right] d\eta \, d\zeta \tag{19}$$

Temperature flux $$H = \int_A \Delta T \, u \, d\eta \, d\zeta \tag{20}$$

where A is the cross-sectional area of the jet.

The continuity Equation, 13, uses the entrainment concept proposed by Morton, Taylor and Turner (6) which relates the normal velocities at the jet boundary to the centerline velocity by means of a proportionality coefficient. α_o is the constant coefficient for lateral entrainment, and α_v is the variable coefficient for vertical entrainment which is a function of the local buoyant damping of turbulent entrainment, characterized by F_L, so that

$$\alpha_v = \alpha_o f(F_L)$$

as indicated by the data of Ellison and Turner (7). Equation 14 expresses the balance between longitudinal momentum and buoyant pressure force. The heat conservation equation, 15, allows for excess heat decay to the atmosphere. The jet spreading equation, 16, represents the realization that the buoyancy of the jet causes spreading of the jet width $(\frac{db}{dx})_B$, in addition to the usual non-buoyant turbulent spreading, ε. Closure of the equations requires specification of $(db/dx)_B$ through the use of the lateral momentum equation. Different hypotheses are possible:

i) Stolzenbach and Harleman (2,3) assume the local lateral velocity, v, to be proportional to the local lateral density gradient, $\partial\Delta\rho/\partial y$, and the local longitudinal velocity, u, the proportionality constant being equal to $(db/dx)_B$. This specification allows integration of Eq. 11 and thus the jet spreading Eq. 16 is in fact replaced by a more complex equation.

ii) Prych (4) and Stefan and Vaidyaraman (5) explicitly solve the lateral momentum equation under simplifying assumptions which reduce the problem to one of unsteady buoyant spread as shown in Fig. 2.

The governing equations for unsteady spreading have been discussed by Koh and Fan (8). In general, three flow regimes are present, a time-dependent inertial regime, a convective inertial regime and a frictional regime. The first regime is important only for small initial times, the latter one only for large widths

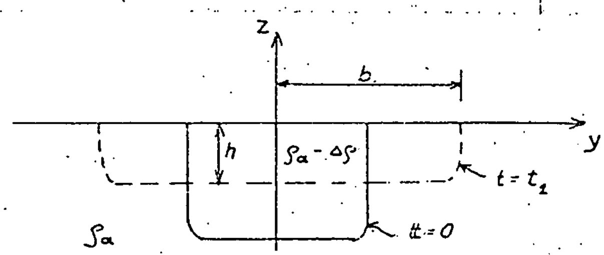


Fig. 2: Unsteady One-Dimensional Buoyant Spread with Constant Volume

$$b \gtrsim h/\lambda_i \qquad (21)$$

where λ_i is an interfacial friction factor defined by $\lambda_i = -\overline{v'w'}/v^2$. In the intermediate range the instantaneous front velocity, v_f, is given by

$$v_f = \frac{db}{dt} = c_4(\Delta\rho\, gh)^{1/2} \qquad (22)$$

where c_4 is a drag coefficient of order unity. This time-dependent spread may be translated to x-dependence by using a transformation $dt = dx/u_p$, where u_p is some representative longitudinal velocity, such as average or centerline velocity. Thus Stefan and Vaidyaraman derive

$$\left(\frac{db}{dx}\right)_B = c_4/F_L \tag{23}$$

and Prych, somewhat differently,

$$\left(\frac{db}{dx}\right)_B = c_4/\sqrt{F_L^{\,2}-1} \tag{24}$$

due to the argument that the spreading should occur normal to the jet boundary instead of normal to the jet axis.

The success of any spreading assumption, such as ~~Stolzenbach and Harleman's~~ approach or ~~Eqs. 23 or 24, has to~~ be seen by comparing to experimental data. Practically, any assumption leads to an overprediction of jet spread whenever F_L becomes small. As an example, see Fig. 3: good agreement is obtained for centerline temperatures and jet thickness while the jet width prediction strongly deviates from data at a certain downstream distance. This overprediction is probably due to two facts: 1) The retention of lateral jet-like profiles at small F_L, 2) The neglecting of lateral shear, $\overline{v'w'}$, in the lateral momentum Eq. 11, which is done by all authors. In fact, most experimental data indicate that disagreement between theory and experiment arises at a distance when the local ratio $b/h \sim 100$, which is also indicated from Eq. 21, as λ_i is in the order of 0.01.

Conclusions: The vertical buoyant acceleration coupled with the density discontinuity of the free surface gives rise to horizontal pressure forces in surface buoyant jets. These forces exert a distorting effect on the jet and thus theoretically restrict the applicability of boundary-layer type jet analysis with the convenient integral technique. However, it has been shown that modifications of the classical jet diffusion equations, such as Eqs. 13 to 16, adequately describe the transition in surface jets from more non-buoyant to more buoyant behaviour. Yet the limiting case of strongly buoyant behaviour is not predicted with required accuracy. It appears, however, that some additional refinement is possible through the explicit incorporation of a lateral shear term in the spreading equation.

Another restriction is given as follows: in the scaling process it has been assumed that the depth of the buoyant jet

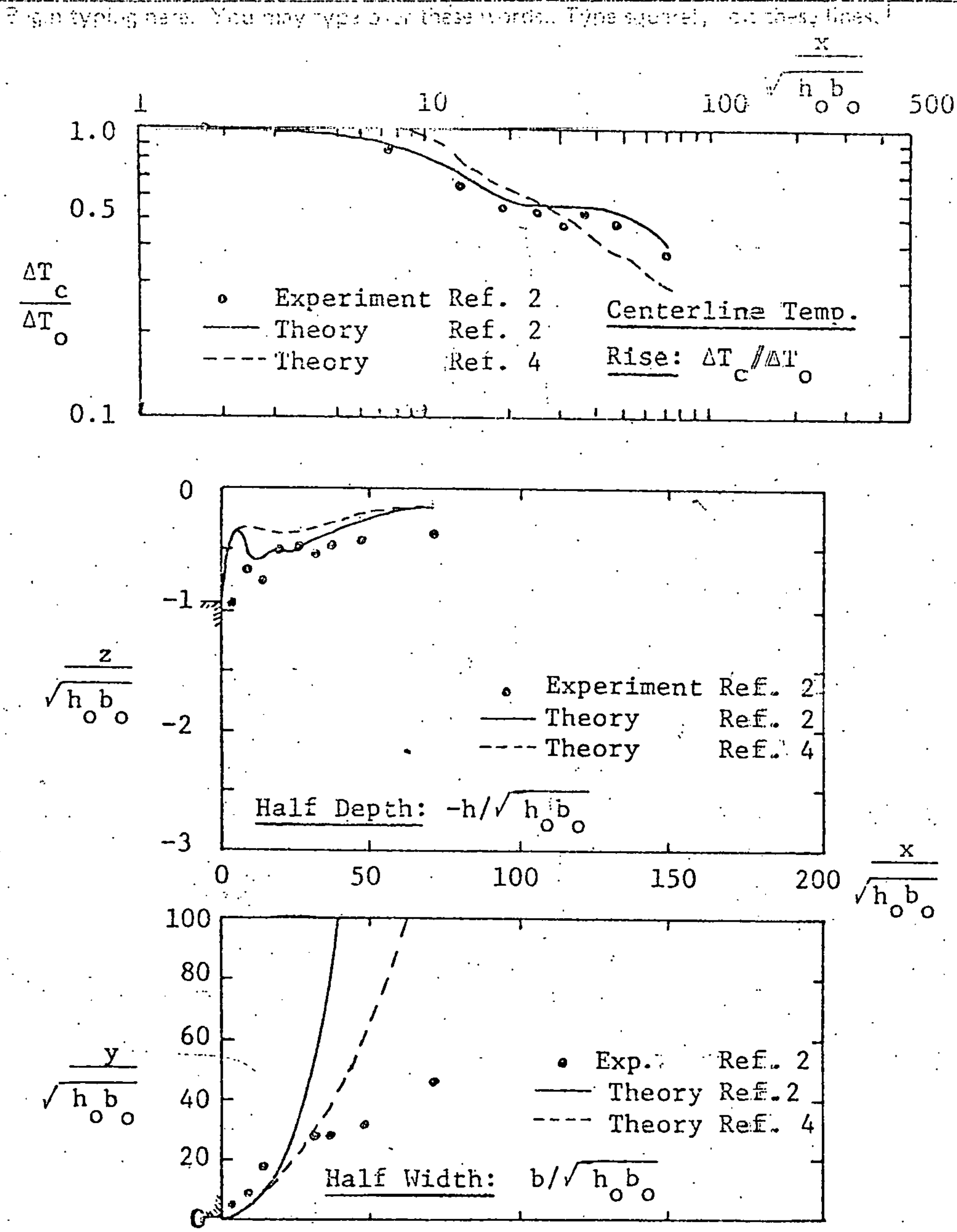


Fig. 3: Comparison Between Experiments and Theoretical Predictions for a Surface Buoyant Jet

$F_o = U_o\left(\frac{\Delta\rho_o}{\rho_a} g\, h_o\right)^{-1/2}$, $h_o/b_o = 0.87$ and

$k/U_o = 6.2 \times 10^{-5}$ (adapted from Ref. 2) is only slowly varying. This is permissible for the gradual transition from large to small F_L. For discharges with initial small F_L, however, abrupt changes in jet depth in conjunction with strong vertical accelerations can occur, in particular for jets with high aspect ratio, i.e., narrow and deep exit discharges. In this case, neglecting the dynamic pressure term and the vertical momentum equation is no longer justified and the integral approach appears not feasible.

Buoyant Submerged Jets in Shallow Water

Recirculation phenomena are of frequent occurrence for jets discharged in confined surroundings. Examples include the homogeneous (non-buoyant) jet impinging on a wall. Pressure gradients which exist in the fluid are the agents responsible for such recirculation. Whenever recirculation into the jet proper occurs simple jet analyses cannot be used to predict jet behavior.

Submerged buoyant jets rise to the water surface under the influence of buoyancy and as a function of the initial discharge angle. Upon impingement on the free surface the flow spreads horizontally. The tendency for recirculation in the vertical direction is inhibited due to the stabilizing influence of buoyancy. This qualitative difference between non-recirculating ("stable") and recirculating ("unstable") jet discharges is shown in Fig. 4 for the case of a vertical discharge.

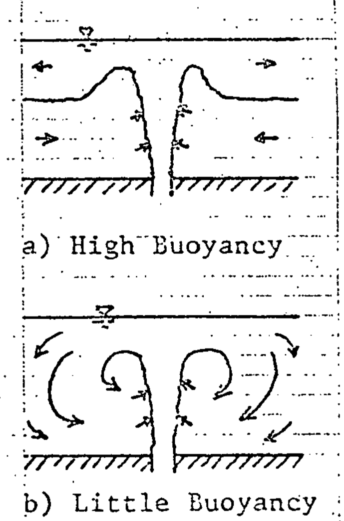


Fig. 4: Effect of Buoyancy on the Stability of the Jet Discharge

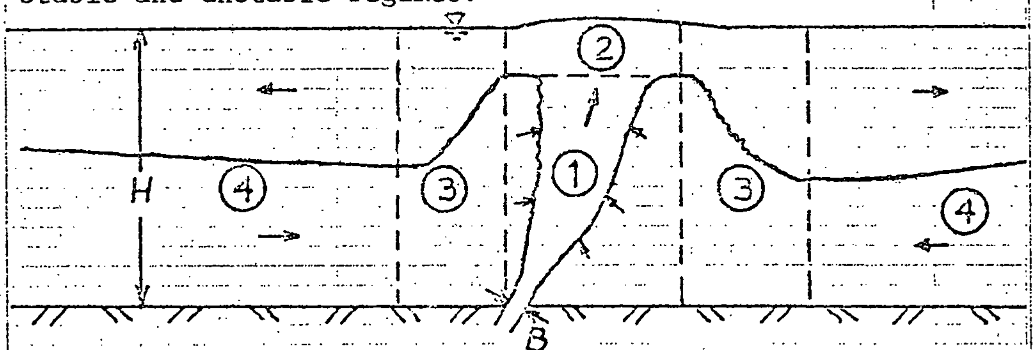
The question of near-field stability as a function of jet parameters is important for both three-dimensional (round jet) and two-dimensional slot discharges. To date, no analysis has been proposed for the three-dimensional case.

The two-dimensional case has been analysed by Jirka and Harleman (9) and is of particular interest in the prediction of submerged multiport diffusers. A multiport diffuser is essentially a pipeline, laid at the bottom of the receiving water, with many nozzles of diameter, D, attached at a regular spacing, ℓ. The individual round jets emanating from the nozzles interfere after a short distance and form a two-dimensional jet zone. It has been shown conclusively (9) that the jet parameters in this two-dimensional zone are equal to those of an "equivalent slot diffuser" with slot width

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

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 designs.

Stability analysis: The distinct flow regions which can be observed in an experimental investigation of a two-dimensional buoyant jet in finite water depth are shown in Figure 5 for the case of a stable near field. These regions are (i) a buoyant jet region, (ii) a surface impingement region, (iii) an internal hydraulic jump, and (iv) a stratified counterflow region. These regions, each with different hydrodynamic properties allowing simplifying approximations, can be analysed separately. Successive matching of the individual solutions yields a description of the total flow field. Objective of the analysis is to determine the limiting condition of a stable flow field, that is, the criterion line between stable and unstable regimes.



1: Buoyant Jet Region | 2: Surface Impingement Region

3: Hydraulic Jump Region | 4: Stratified Counterflow Region

Fig. 5: Vertical Structure of a Two-Dimensional Slot Jet in Finite Water Depth (stable near-field)

Stability is primarily dependent on the regions (i), (ii) and (iii). Dimensional analysis of the problem gives the following governing parameters:

Slot densimetric Froude number: $F_s = U_o \left(\frac{\Delta\rho_o}{\rho_a} gB\right)^{-1/2}$

Relative water depth: H/B

Angle of discharge: θ_o

where $\Delta\rho_o$ = initial density difference and H = water depth.

(i) Buoyant Jet: A hydrostatic pressure distribution is assumed in this region. This is tantamount to the assumption that the pressure disturbance due to the rise in surface elevation as a result of impingement is limited to the impingement region. This assumption is essentially verified by experiments. A buoyant jet analysis, utilizing the entrainment concept, is performed to give predictions of jet dilutions and trajectories.

(ii) Surface Impingement: The surface impingement region provides the transition between the jet flow, with a strong vertical component, and the horizontal spreading motion. The process is a complex flow phenomenon, which is most conveniently analysed by means of a control volume approach, using a continuity equation, a horizontal momentum equation and two energy equations (account is being taken for the energy loss in the flow transformation). Results of the analysis give the thickness of spreading layer, h_1, and thus the elevation up to which effective jet entrainment occurs. Furthermore, the dynamic characteristics of the spreading layer, represented by a densimetric Froude number

$$F_1 = u_1\left(\frac{\Delta\rho}{\rho_a} g h_1\right)^{-1/2} \tag{25}$$

where u_1 = layer velocity, $\Delta\rho$ = relative density difference between upper and lower layer, can be calculated. Fig. 6 summarizes the results over the parameter range for the case of vertical jet discharge. The layer thickness is about 1/6 of the total water depth, only weakly dependent on F_s and H/B. The Froude number F_1 is strongly dependent on F_s and H/B: high values of F_1 persist in the shallow water - high F_s range.

(iii) Internal Hydraulic Jump: Experimental observations indicate that following the surface impingement the thickness of the surface layer suddenly increases in form of an internal hydraulic jump. In analogy to free surface flow a hydraulic jump provides a transition between a supercritical upstream section and a subcritical downstream section. The criticality of a stratified system is defined by

$$F_1^2 + F_2^2 = 1 \qquad (26)$$

following Schijf and Schönfeld (10), where F_2 is the lower layer densimetric Froude number analogous to Eq. 25. A supercritical stratified flow section is then characterized by

$$F_1^2 + F_2^2 > 1 \qquad (27)$$

and a subcritical one by

$$F_1^2 + F_2^2 < 1 \qquad (28)$$

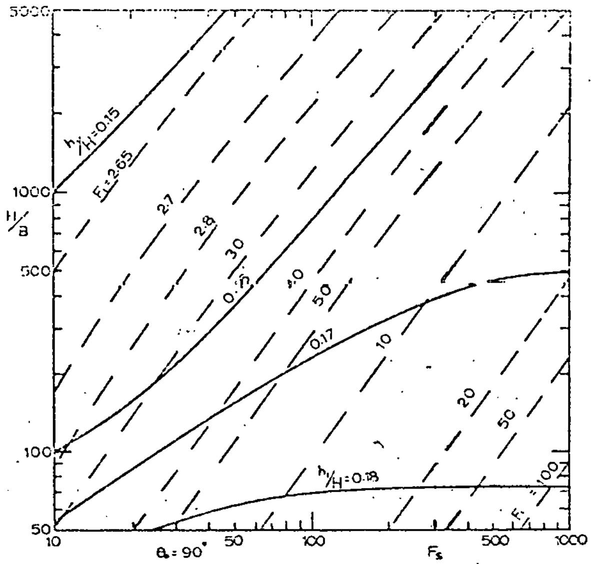


Fig. 6: Thickness h_1/H and Densimetric Froude Number F_1 of the Surface Impingement Layer (Vertical Discharge)

That the flow section following the surface impingement is indeed supercritical can be shown by using the values for F_1 from Fig. 6: inequality 27 is satisfied over the whole parameter range.

The governing equations for internal hydraulic jumps have been derived by Yih and Guha (11). A simplified solution for small density differences has been obtained by Jirka and Harleman (9). The equations indicate that for certain upstream conditions no solution is possible, that is, a stable subcritical downstream condition does not exist. In general, these upstream conditions are characterized by strong supercriticality, namely high values for $(F_1^2 + F_2^2)$. The criterion line between those conditions where an internal hydraulic jump is possible and those where no jump is possible and hence recirculation occurs is shown in Fig. 7 for various discharge angles, Θ_o. The shallow water - high F_s is generally characterized by near-field instability. Furthermore, for decreasing discharge angles (more horizontal discharge) the instability is enhanced.

Dilution prediction: The variable of major importance in the evaluation of submerged discharges is the resulting surface

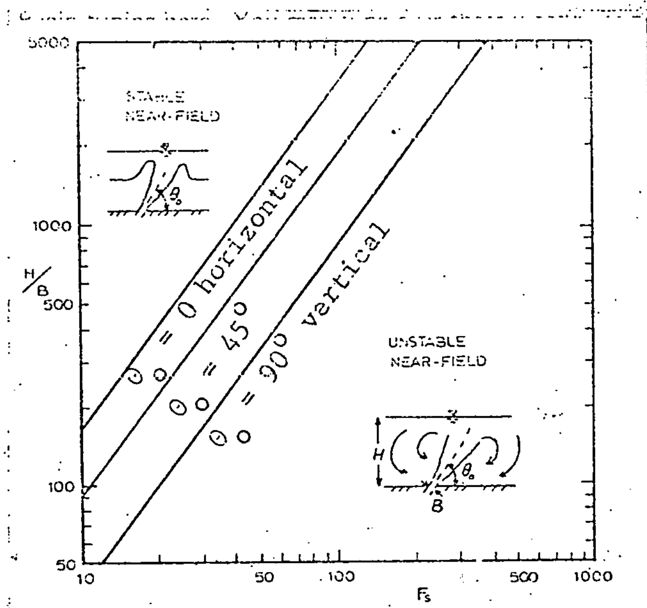


Fig. 7: Stability of the Near-Field Zone as a Function of Discharge angle, Θ_o

dilution, S_s. For the case of a stable near-field S_s can be directly obtained from the buoyant jet analysis, if account is being taken for the thickness of the impingement layer.

Whenever the near-field is unstable and recirculation into the jet zone occurs, then a simple buoyant jet ~~analysis is not valid.~~ In this case the near-field dilution is directly dependent on the stratified counterflow system in the far-field. This counterflow presents a balance between buoyancy forces and frictional forces (dependent on geometry, boundary and interfacial roughness). A variety of three-dimensional flow conditions can result in the far-field, dependent on geometry and horizontal momentum input by the diffuser. Values for S_s, evaluated through analysis of these different far-field conditions have been obtained by Jirka and Harleman (9). It is also shown in that reference that thermal diffusers for cooling water discharge are usually characterized by an unstable near-field.

Conclusions

While jet models certainly present one of the most important techniques for thermal analysis, their application is limited whenever some of the underlying assumptions are violated. In case of buoyant surface jet models buoyancy will ultimately cause strong lateral spreading. Retention of jet-like profiles and neglecting of lateral shear leads to inaccuracies in the solution in comparison to available data. While existing models are capable to predict the transition from initial momentum-dominated behavior to more buoyant behavior, the limiting case of strongly buoyant spreading is not adequately predicted.

Pressure disturbances occur whenever a submerged buoyant jet impinges on the free surface. When the jet is strongly buoyant, as indicated by a low densimetric Froude number and high relative water depth, then the buoyancy is sufficient to prevent any vertical recirculation into the jet zone and buoyant jet models are adequate for predictive purposes. Yet in case of recirculation, which is typical for cooling water discharges, different approaches have to be used for prediction, such as outlined by Jirka and Harleman (9).

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