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The Johns Hopkins University

TECHNICAL REPORT NO. 75

Topical Report Under
The U.S. Atomic Energy Commission
Contract No. AT(30-1)-3109
and Contract No. AT(11-1)-3062

AEC Document No. COO-3062-3

DESIGN AND SITING CRITERIA
FOR ONCE-THROUGH COOLING
SYSTEMS BASED ON A FIRST-
ORDER THERMAL PLUME MODEL

D.W. Pritchard
and H.H. Carter

Reference 72-6

April 1972

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**D. W. Pritchard
Director**

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ABSTRACT

The processes which control the distribution of excess temperature in the heated plume resulting from the discharge of a heated effluent into a natural waterbody are discussed, and the relative importance of each process for various plant design and environmental conditions is assessed. The distributions of excess temperature, and the time-temperature history of exposure of organisms entrained into the thermal plume resulting from the discharge of condenser cooling water from a 1000 MWE nuclear power plant, are given for several alternate sets of design criteria for the discharge structure. It is shown that, in order to minimize adverse effects on natural water bodies used to supply condenser cooling water flow for once-through cooling systems, the power plant should be located on a waterway having large dilution capacity, and the discharge structure should be designed to promote rapid dilution of the heated effluent by the cooler receiving waters. Intake structures should be designed to provide for low intake velocities, and to provide an effective means of diverting fish and other aquatic organisms away from the intake. Time of travel of the condenser cooling water flow from the condensers to the point of discharge should be kept small, in order to minimize the time of exposure of entrained organisms to the maximum temperature rise.

I. Introduction

The purposes of this report are: (1) to describe the processes which control the distribution of excess temperature in the heated plume resulting from the discharge of a heated effluent into a natural water body; (2) to assess the relative importance of each process for various plant design and environmental conditions; (3) to discuss the probable effect of various features of plant design on the aquatic biota; (4) to use the results of the foregoing descriptions, assessments and discussions in setting forth design and siting criteria for steam-electric power stations using once-through cooling systems which will minimize environmental effects of plant operations; and (5) finally, to give first order estimates of the relative size of the various sources of man-made inputs to the heat budget of the earth surface boundary layer, and to briefly assess the relative environmental impact of various alternate means of transfer of the excess rejected heat from electric power production to the environment. In order to attain these objectives use is made of a first-order thermal plume model.

II. The Fate of Excess Heat Discharged into a Natural Water Body

A. A General Discussion

Two categories of processes control the distribution of excess temperature in the waters of a lake, an estuary, or off a coastline into which the condenser cooling water from an electrical generating plant is discharged. The ultimate sink for all the excess heat thus discharged is the atmosphere. Thus one of the two categories of processes is the loss of heat from the water surface to the overlying air. The second category is dilution, which results both from the process of momentum entrainment and the process of natural mixing, of the heated effluent by the cooler receiving waters. Despite the fact that the ultimate fate of all the excess heat discharged is a transfer from the water to the

atmosphere by surface cooling, it has been clearly demonstrated that in the case of discharge of heated effluent into a relatively large body of water (that is, a body of water having a surface area large compared to that required for transfer of the excess heat to the atmosphere even at very small surface excess temperatures), or a water body having a natural renewal rate due to inflow of "new" water which is large compared to the volume rate of discharge of heated effluent, that the distribution of excess temperatures, in that segment of the thermal plume in which the excess temperatures exceed approximately one-tenth the initial excess temperature, is determined primarily by the processes of dilution. Surface cooling plays a dominant role only in areas having relatively small excess temperatures. Example computations will be given later in this paper which will clearly demonstrate this statement.

Each of the dominant processes involved in controlling the distribution of excess temperature in a thermal plume resulting from the discharge of condenser cooling water into a natural water body will now be examined.

B. Surface Cooling

Consider an increment volume of water introduced into a natural water body with the condenser cooling water discharge. This incremental volume is considered to extend from the surface to some depth D_h , and to have unit surface area. The subject volume enters the waterway with a certain amount of excess heat, that is, the heat rejected at the condensers, and consequently has an initial excess temperature, which will be designated as θ_o .

As the subject volume moves with the discharged thermal plume, its excess temperature will decrease due to horizontal mixing with cooler water through the side boundaries of the volume, and due to vertical mixing through the bottom boundary. Heat will also be lost to the atmosphere across the air-water interface. For the moment

consider only this latter process, surface cooling, neglecting any dilution of the excess heat by entrainment or turbulent diffusion across the water faces of the volume.

The quantitative expression for the rate of loss of excess heat is obtained by taking the difference between the heat budget of the subject volume, and the heat budget which would apply to the volume if no excess heat had been added. That is, the time rate of change of heat content of the volume under natural conditions (assuming no heat rejected at the condensers) is given by

$$(1) \quad \frac{\partial h}{\partial t} = q_s - q_{sr} + q_a - q_{ar} - q_b - q_e - q_c - q_{we}$$

where: h = heat content of the volume
 q_s = solar radiation incident on the water surface
 q_{sr} = reflected solar radiation at the water surface
 q_a = incoming long-wave radiation from atmosphere incident on water surface
 q_{ar} = reflected long-wave radiation at water surface
 q_b = long-wave back radiation from water surface
 q_e = rate at which heat is utilized for evaporation
 q_c = rate at which heat is conducted from the water surface as sensible heat
 q_{we} = rate at which heat is removed from the water body in the evaporated water.

The heat budget for the subject volume under conditions of added excess heat can be written as

$$(2) \quad \frac{\partial h'}{\partial t} = q_s - q_{sr} + q_a - q_{ar} - q_b' - q_e' - q_c' - q_{we}'$$

where the primed terms are those which would be changed by the presence of excess heat. Note that the terms involving solar radiation and long-wave radiation from the atmosphere would be the same in both cases. Subtracting equation (1) from equation (2), gives

$$(3) \quad \frac{\partial}{\partial t} (\Delta h) = - \Delta q_b - \Delta q_e - \Delta q_c$$

$$\text{where } \Delta h \equiv h' - h \quad ; \quad \Delta q_b \equiv q_b' - q_b$$

$$\Delta q_c \equiv q_e' - q_e \quad ; \quad \Delta q_c' \equiv q_c' - q_c$$

and where $q_{we}' - q_{we}$ can be shown to be negligible.

Now Δh is the excess heat content of the subject volume, and the left side of equation (3) represents the rate of loss of excess heat from the volume due to surface cooling. The three terms on the right side of equation (3) represent (a) the difference between the long-wave back radiation from the water surface for the heated and unheated situations, (b) the difference between the evaporative heat losses for the heated and unheated situations, and (c) the difference between the conductive heat losses for the heated and unheated situations.

The long-wave back radiation from the water surface is a known function of surface temperature (i. e., the Stefan-Boltzmann law). Investigations such as the Lake Heffner Study (1) have provided quite reliable relationships giving the evaporation from a lake surface as a function of the wind velocity and the difference between the saturated vapor pressure at the temperature of the water surface and the actual vapor pressure in the air over the water. Expressions for evaporation from a natural water body, such as those developed in the Lake Heffner study, must be modified for the case of evaporation from a water surface at temperatures above the natural temperature to provide for the increased atmospheric convection over such a heated surface. The resulting relationship provides for evaporation at zero wind velocity. The conductive heat exchange at the surface can be shown to be related to the evaporative loss through a function called the Bowen's Ratio. Thus the terms on the right side of equation (3) can be computed given water temperatures and atmospheric parameters. It should be noted that when the difference between the evaporation under heated conditions and under unheated conditions is taken,

the actual vapor pressure in the air over the water drops out of the relationship. Thus the evaporative loss of excess heat is independent of relative humidity, depending only on the difference between the saturated vapor pressures at the temperatures of the water surface for the heated and unheated situations.

Over the normal range of natural water temperatures (i. e., 40°F to 80°F) and of excess temperatures (i. e., 0°F to 20°F), the expressions for the three terms on the right side of equation (3) can be shown to be satisfactorily represented by a simple linear relationship having the form:

$$(4) \quad \frac{\partial}{\partial t} (\Delta h) = - \mu \theta$$

where $\theta \equiv (T_h - T_n)$ is the excess temperature of the subject volume (T_h designates the surface temperature for the heated situation and T_n the surface temperature for the natural, or unheated situation), and μ is a slowly varying coefficient (the surface cooling coefficient), which depends primarily on wind velocity, secondarily on natural water temperature, and within the range of excess temperatures normally encountered, only to a third order on excess temperature. Now the excess heat content of the subject volume at any time is given by

$$(5) \quad \Delta h = \rho c_p D_h \theta$$

where ρ is the water density, c_p the specific heat, and D_h is, as noted earlier, the vertical thickness of the layer containing the excess heat. Substituting (5) into (4) gives

$$(6) \quad \frac{\partial \theta}{\partial t} = - \frac{\mu \theta}{\rho c_p D_h}$$

Table 1 gives μ the surface cooling coefficient, expressed in $\text{BTU} \cdot \text{ft}^{-2} \cdot (^\circ\text{F})^{-1} \cdot \text{hr}^{-1}$, as a function of wind velocity W , natural temperature T_n and excess temperature θ . It can be seen that μ depends primarily on W and T_n , and hence equation (6) can be integrated to produce the following expression:

$$(7) \quad \theta_\tau = \theta_o \exp \left\{ - \frac{\mu \tau}{\rho c_p D_h} \right\}$$

where θ_τ is the excess temperature after some time interval τ , and θ_o is the initial excess temperature. In the absence of any mixing with the receiving waters, the surface area within any isoline of constant excess temperature, A_θ , is related to the time interval since discharge, τ , the depth interval containing the excess heat, D_h , and the volume rate of discharge of heated condenser cooling water Q_c , such that

$$\tau/D_h = A_\theta/Q_c$$

Hence equation (7) can be written

$$(8) \quad \theta_A = \theta_o \exp \left\{ - \frac{\mu A_\theta}{\rho c_p Q_c} \right\}$$

Or, solving for A_θ , the surface area having excess temperatures equal to or greater than θ_A , we have

$$(9) \quad A_\theta = \frac{\rho c_p Q_c}{\mu} \ln \frac{\theta_o}{\theta_A} = \frac{Q_h}{\theta_o \mu} \ln \frac{\theta_o}{\theta_A}$$

where Q_h is the rate of rejection of heat at the condensers, and is given by $Q_h = \rho c_p \theta_o Q_c$.

Table 1

The Surface Cooling Coefficient, μ ($\text{BTU} \cdot \text{ft}^{-2} \cdot (^\circ\text{F})^{-1} \cdot \text{hr}^{-1}$) as a Function of the Wind Speed, W (mph), the Natural Surface Water Temperature, T_n ($^\circ\text{F}$), and the Excess Temperature, θ ($^\circ\text{F}$)

W	θ				
	2°	6°	10°	14°	18°
(For $T_n = 40^\circ\text{F}$)					
2	1.39	1.51	1.65	1.83	2.07
4	1.90	2.05	2.21	2.42	2.68
6	2.42	2.58	2.78	3.00	3.29
8	2.94	3.12	3.33	3.59	3.89
10	3.46	3.66	3.90	4.18	4.50
(For $T_n = 60^\circ\text{F}$)					
2	1.73	1.87	2.05	2.30	2.61
4	2.47	2.69	2.88	3.17	3.52
6	3.23	3.45	3.70	4.02	4.42
8	3.98	4.23	4.53	4.89	5.32
10	4.73	5.02	5.36	5.75	6.21
(For $T_n = 80^\circ\text{F}$)					
2	2.26	2.44	2.68	3.03	3.47
4	3.43	3.66	3.97	4.38	4.88
6	4.59	4.90	5.27	5.74	6.31
8	5.77	6.11	6.57	7.08	7.71
10	6.93	7.37	7.86	8.45	9.13

Equation (9) can be used to compute the area inside a given excess temperature isotherm for the case of surface cooling only - - i. e., no dilution by momentum jet entrainment or by natural processes of turbulent diffusion.

Publications containing background material pertinent to the subject of this section are included in the Reference List given at the end of this report. Of particular pertinence are Reference Numbers 1, 2, 3, 4, 5 and 6.

From a strictly theoretical standpoint, the treatment given above applies only when the natural temperature would have been equal to the equilibrium temperature, since cooling rate is actually dependent on the difference between the temperature under heated conditions and the equilibrium temperature. From a practical standpoint, during the periods of maximum temperatures in summer and of minimum temperatures in winter, the natural temperatures would be very nearly equal to the equilibrium temperatures, and the equations given above should be applicable. During spring, when the natural temperatures are increasing rapidly, the equilibrium temperature is higher than the natural temperature. The driving temperature difference for a heated element will therefore be less than the excess temperature, and cooling will be at a lower rate than indicated by the equations given above. In autumn, when the natural temperature is decreasing rapidly, the equilibrium temperature is lower than the natural temperature. The driving temperature difference for a heated effluent would in this case be greater than the excess temperature, and cooling would be at a greater rate than indicated by the equations given above. The practical consequence of these effects is a slight shift in time of the rising temperature vs. time curve in spring, and in the falling temperature vs. time curve in fall. As stated earlier, and as will be demonstrated below, the fact that dilution is the dominant process in determining the distribution of significant values of excess temperatures further contributes to making the difference between the natural temperature and the equilibrium temperature of academic rather than practical importance.

In recent years several investigators have argued that the surface cooling coefficient in large natural water bodies is significantly larger than would be predicted from the semi-theoretical, semi-empirical development outlined above. In particular Edinger and Geyer, in a study sponsored by the Edison Institute (Reference 3), presented data for a small flow-through cooling pond which receives the heated discharge from a power plant and discharges the partially cooled water to an adjacent tidal waterway. This study appeared to indicate cooling coefficients several times larger than those computed from a theoretical development similar to that given above. However, two factors make suspect the conclusions arrived at in the subject report. First, the mean temperature in the pond differed from the temperature of the discharge by only about 1°F. Practical difficulties in determining the difference between the discharge temperature and the mean pond temperature could contribute to significant errors in the computed surface cooling coefficient. Of more significance is the fact that the discharge weir between the pond and the adjacent tidal waterway was one foot below mean low tide level, thus allowing some of the cooler water from the adjacent tidal waterway to enter the pond at certain stages of the tide. Consequently a part of the observed reduction in the excess temperature within the pond resulted from dilution, and not from surface cooling.

Pertinent to this discussion are the results given in a later report put out under this same program sponsored by the Edison Institute (Reference 6). That study shows that empirically determined cooling coefficients, for cases where dilution can be definitely eliminated as a cause for decreases in excess temperature, are related to wind velocity and temperature in very nearly the same manner given earlier in this paper for wind velocities of 5 to 10 mph. For wind velocities of less than 4 mph, the empirically determined cooling coefficients were somewhat higher than those given here.

Our own studies of the temperature distribution in natural water bodies adjacent to thermal discharges have shown that, because of large natural fluctuations in the spacial and temporal distribution of ambient temperature, it is extremely difficult to establish the probable value of the natural temperature which would have prevailed had there been no discharge of heated effluent. The reason that some investigators have concluded that field studies of the temperature distribution resulting from the discharge of heated effluent to natural water bodies indicate much larger cooling coefficients than theoretically predicted, or than have been shown to apply to cooling ponds, is the failure of these investigators to recognize the importance of dilution, and the fact that most of the excess heat is lost to the atmosphere from large areas of very low excess temperature.

C. Horizontal Turbulent Diffusion

The vertical stability at the bottom of the heated layer, resulting from the warm thermal plume spreading over the cooler receiving waters, will inhibit vertical mixing.

The vertical thickness of the thermal plume in the "near field" (i. e., the segment of the plume in which entrainment dominates the dilution process) depends upon the combined effects of inertia and buoyancy. In the "far field", outside the segment of the plume in which the dilution of the heated effluent is controlled primarily by entrainment, the vertical penetration of temperatures above ambient will also depend on the natural processes of vertical mixing. In many cases the depth of water in the receiving waterway will restrict vertical spread of the heated water and any natural stable vertical density stratification will also limit the vertical thickness of the thermal plume.

As a result of all these factors, dilution of the plume beyond an initial transition region will be controlled primarily by horizontal

entrainment and horizontal turbulent diffusion. A number of investigators have described the theoretical and observational information on horizontal diffusion from a two-dimensional point source (i. e., a vertical line source) in natural water bodies (7), (8), (9), (10). Consider a tracer material or a contaminant introduced as a continuous vertical line source of vertical thickness D , within a water layer of the same thickness bounded at the top and bottom by interfaces which prohibit or inhibit vertical motion and mixing (for example the water surface and the bottom, or the water surface and the thermocline), unbounded laterally, and in a uniform horizontal current field. Most theories and experiments show that the peak concentration along the centerline of the plume of tracer which develops in the down-current direction will decrease as the inverse first power of distance from the source. The exact form of the transverse distribution across the plume as predicted by the several theories differs somewhat depending on which particular theoretical expression is employed. However, all theories predict that the lateral distribution in the plume will have a bell-like form, and experimental evidence indicates that the average lateral distribution in the plume is approximately Gaussian in form.

When the source is a relatively large continuous discharge, such as the discharge of condenser cooling water, natural turbulent diffusion initially acts only on the edges of the plume, and has little effect on decreasing the concentration of contaminant near the center of the plume until the transverse distribution has been altered from an initial relatively flat-topped form to a bell-shaped form. Thus the rate of decrease of the maximum concentration with distance along the centerline of the plume due to natural turbulent diffusion is, in the case of a large volume rate of discharge, very small for a considerable distance from the source. At sufficiently large distances from the source, however, the concentration distribution will approach that for a two-dimensional point source, and the maximum concentration will thereafter decrease as the inverse first power of distance.

Thus the natural processes of turbulent diffusion are ineffective in producing a rapid reduction in the excess temperature with distance in the vicinity of the point of discharge. In order to promote rapid reduction in excess temperature with distance close to the point of discharge, the discharge structure must be designed to provide for a high degree of mechanical dilution.

The above discussion applies to the processes of small scale and intermediate scale turbulent-shear diffusion. The excess temperature distribution in the thermal plume will also depend upon the approximate steady state excess temperature of the receiving waters being entrained into and mixed with the plume, which might be appropriately called the "entrainment" excess temperature. This entrainment excess temperature depends upon the rate of supply of new dilution water to the segment of the waterway containing the definable thermal plume from adjacent segments of the waterway, due to large scale exchange processes, as well as on the rate of loss of excess heat from the surface to the atmosphere. For example, in a tidal estuary, the new water available for dilution of the thermal plume is supplied by both the river inflow to the estuary and by seawater entering the mouth of the estuary, and by large scale tidal exchange processes between adjacent segments of the estuary.

D. Momentum Jet Entrainment

1. Horizontal Surface Jet in Absence of Longshore Currents

A number of experimental and theoretical studies have been conducted in recent years on the entrainment of diluting waters into both momentum jets and buoyant jets, particularly in reference to the design of effective submerged offshore diffusers for municipal sewage discharges into coastal waters (11), (12). A few studies have also involved the horizontal discharge of a heated effluent as a surface momentum jet (13), (14). These latter studies have shown that in a receiving waterway of infinite depth momentum entrainment occurs both horizontally through the

sides of the thermal plume and vertically through the bottom of the plume. As a result of this three-dimensional entrainment, the excess temperature along the axis of the plume resulting from the horizontal discharge of a heated effluent through a circular or rectangular-shaped orifice varies as the inverse first power of distance from the orifice. In a natural water body, the depth is finite, and in many cases will limit vertical entrainment. Usually, at least in the warmer period of the year, the water body will have a natural stable vertical density gradient. The combination of this natural vertical stability with the added buoyant stability at the bottom face of the thermal plume will also significantly limit vertical entrainment. On the basis of a number of studies of actual temperature distributions in waterways receiving heated discharges, we conclude that a relatively simple first-order model involving only horizontal entrainment can be used to indicate the criteria for design and siting of once-through cooling systems in order to minimize adverse environmental impact.

The following briefly describes such a first-order theory for the entrainment of diluting waters into a horizontal surface jet.

Consider the discharge of a heated effluent from a conduit extending into a large natural water body, and directed approximately at right angles to the shoreline. The width of the discharge orifice is designated by b_o , and the depth by h_o . It is assumed that the natural water depths are at least as large as h_o at the point of discharge, and increase with distance along the direction of discharge of the thermal plume. It is also assumed that the water depths are sufficiently great so that there is no loss of momentum of the jet due to bottom friction.

The initial velocity of the jet at the point of discharge will be directed at right angles to the shoreline and will have an initial speed given by

$$(10) \quad u_j = Q_c / b_o h_o$$

where Q_c is the volume rate of flow of the condenser cooling water. Any natural flow in the receiving water body will usually be parallel to the shoreline. Thus the jet enters the receiving water body with an excess

momentum directed at right angles to the shoreline, and with zero momentum directed parallel to shore. The jet must lose its excess momentum in the direction perpendicular to the shoreline and gain momentum in the direction parallel to the shoreline (a) by entraining water of ambient momentum, and (b) as a result of the pressure distribution set up along the upstream boundary of the bending jet.

If the longshore ambient current is sufficiently large compared to the velocity of discharge in the jet, the plume would be bent so far over that entrainment on the inshore side of the plume would be reduced, and the dilution rate would be correspondingly reduced. The modifications to the theory which must be made in the case of a relatively strong longshore current will be given later. For the present consider the case for which the longshore currents can be neglected.

If it is assumed, for purposes of keeping the mathematics of the problem tractable, that the distribution of both momentum and any other identifiable property (i. e., excess temperature) maintains a transverse distribution having a "top-hat" form, which expands linearly with distance it can be shown from the requirements for the integral conservation of momentum and of any other identifiable conservative property of the jet, that along the centerline of the thermal plume,

$$(11) \quad \frac{\theta_{\xi}}{\theta_0} = \left\{ \frac{\xi_{\theta}}{\xi_v} \right\}^{-1/2}$$

where θ_{ξ} is the excess temperature at a distance ξ_{θ} along the centerline of the jet, and θ_0 is the initial excess temperature at the point of discharge. The length ξ_v is the length of the zone of flow establishment and is the distance from the virtual source of the jet, which is located approximately at the discharge orifice, to the point where momentum or material begins to be diluted. That is, there is a certain distance, ξ_v over which no dilution occurs, and hence equation (11) is defined only for $\xi > \xi_v$.

The distance, ξ_v , can be expressed in terms of the width of the orifice as

$$(12) \quad \xi_v = n \cdot b_o$$

substituting equation (12) into (11) gives.

$$(13) \quad \xi_\theta = \left\{ \frac{\theta_o}{\theta} \right\}^2 n b_o$$

Designating the rate of heat rejection at the condensers by Q_h , then

$$(14) \quad \theta_o = \frac{Q_h}{c_p \rho Q_c}$$

and equation (13) can therefore be written in the form

$$\xi_\theta = \left\{ \frac{Q_h}{c_p \rho Q_c} \right\}^2 \times \frac{n b_o}{\theta^2}$$

$$\text{Also, } Q_c = u_j h_o b_o$$

and hence equation (13) can be written in the following alternate forms.

$$(15) \quad \xi_\theta = \left\{ \frac{\theta_o}{\theta} \right\}^2 \cdot \frac{Q_c n}{h_o u_j} = \left\{ \frac{Q_h}{\rho c \theta} \right\}^2 \cdot \frac{n}{Q_c h_o u_j}$$

The mathematical statement of the basic assumption of linear spread is

$$(16) \quad b = b_o \left\{ \frac{\xi}{\xi_v} \right\} = b_o \left\{ \frac{\xi}{n b_o} \right\} = \frac{\xi}{n}$$

Based on measurements of the rate of decrease of excess temperature with distance along the center line of the plume, n has a value close to 6.0, and this value will be used in Equation (13). However, empirical evidence suggests that, on the average, the maximum width of the area contained within any given isotherm of excess temperature is given by $b_{\theta} = \xi_{\theta}/4.0$, and the mean areal width is given by $\overline{b}_{\theta} = \xi_{\theta}/4.65$. These expressions are used in subsequent computations of the general shape and area of specific isotherms, rather than retaining the initial artificial assumption of a "top hat" distribution. Admittedly, these relationships for the width of the thermal plume are based on quite limited data. It is likely that the ratio of width to length of the area contained within a given isotherm in a buoyant plume is not a constant. As more data becomes available we expect to develop more realistic expressions for this parameter.

Keeping in mind that the above treatment deals with the distribution of excess temperature resulting from jet entrainment only, (i. e., surface cooling and natural diffusion are not taken into account), the above expressions can be used to make some general statements about the size of the thermal plume within the region where momentum jet entrainment is the dominant process. Momentum jet entrainment will be the dominant process in controlling the distribution of excess temperature out to greater dilutions, that is, to smaller values of excess temperature, the larger the velocity of the jet at discharge. We have assumed for purposes of the comparisons made later in this paper that this process will be the most important out to a dilution of at least 5:1, that is, to the point where the excess temperature has been reduced to 20% of the excess temperature at discharge. Thus in the following statements derived from the above equations the "size" of the thermal plume is defined by $\theta \geq 0.2 \theta_o$.

(a) If θ_o is held constant (which also requires the ratio Q_h/Q_c to be constant) then the "size" of the thermal plume is directly proportional to b_o .

(b) If b_o is held constant, then the "size" of the thermal plume is directly proportional to θ_o^2 , and hence to $\left\{ Q_h/Q_c \right\}^2$.

(c) If Q_h is held constant, then the "size" of the thermal plume is directly proportional to the ratio b_o/Q_c^2 , or inversely proportional to the product $Q_c h_o u_j$.

2. Horizontal Surface Jet in the Presence of a Longshore Current

Only rather limited studies have been made to date of the trajectory of the jet plume and of dilution due to momentum entrainment in a surface thermal plume discharged into a water body having significant longshore currents. The discussion given here is based on an investigation recently completed by Carter [15]. In that study, Carter showed for cases where the ambient longshore current equaled or exceeded one-tenth of the initial velocity of the jet discharge, the thermal plume would be bent to become parallel to shore, at a distance sufficiently close to shore so that dilution by entrainment on the inshore side of the plume is effectively reduced. Note that Carter did not study situations in which the longshore current velocity was less than 10% of the initial jet velocity. Hence the minimum transverse current which will result in reduction of entrainment on one side of the bent jet plume is not known.

Carter gives a procedure for determining the trajectory of the central axis of the bent jet plume. For our purpose here it is sufficient to summarize Carter's measurements rather than use the theory developed in the referenced paper. Consider a rectilinear coordinate system with origin at the discharge orifice. The y - axis is directed offshore, perpendicular to the shore-line, and the x - axis is directed alongshore, in the direction of flow of the ambient longshore current. In the referenced study [15], Carter investigated only cases in which the discharge orifice was located at the shoreline, and directed at right angles to the shoreline. Further, the shoreline itself was a straight line, and the ambient current field was, in the absence of the jet discharge, directed parallel to shore

(and hence perpendicular to the direction of the jet discharge) and uniform in the y - direction.

Designating:

- y_m = maximum offshore penetration of the axis of the jet plume
- x_m = the distance alongshore at which the maximum offshore penetration of the axis of the jet plume occurs.
- u_j = initial speed of the jet discharge
- u_a = speed of the transverse ambient current
- $R = u_j/u_a$, i.e. the ratio of the initial speed of the jet discharge to the transverse ambient current.

Carter measured the trajectory of the centerline of the bending plume as well as the dilutions as a function of distance along the centerline for various values of the ratio R. Table 2 is a partial summary of these results. It should be noted that Carter also measured the lateral distribution of temperature (i.e. the variation of excess temperature along lines perpendicular to the centerline of the plume) though this data was not reproduced in the subject reference. The lateral data indicated that down-plume from the point at which the trajectory becomes essentially parallel to the shoreline, the excess temperature is very nearly constant from the centerline to the shore.

The trajectory of the bent jet was elliptical between the discharge orifice and the point x_m, y_m , where the trajectory became parallel to the shoreline. From the discharge orifice to the point x_m, y_m , the rate of dilution along the axis of the plume was proportional to the fourth-root of distance. That is, if θ_o is the initial excess temperature of the discharge, and θ_ξ is the excess temperature at a distance along the axis of the plume, ξ , then

$$(17) \quad \theta_\xi / \theta_o = \left\{ \frac{\xi}{mb_o} \right\}^{1/4}$$

Table 2

Values of the relative distance alongshore, x/b_o , offshore, y/b_o , and along the center line, to the point at which the bent jet plume becomes parallel to the shoreline; the dilution, $(\theta/\theta_o)_m$; all as a function of the ratio of the speed of the jet discharge to the speed of the longshore (transverse) ambient current.

<u>R</u>	<u>x_m/b_o</u>	<u>y_m/b_o</u>	<u>ξ_m/b_o</u>	<u>$(\theta/\theta_o)_m$</u>
1	2.15	1.05	2.6	1.00
2	10.78	5.89	12.9	0.67
5	45.70	24.60	58.1	0.46
10	146.80	74.00	194.6	0.34

for ξ greater than mb_0 . Data from the experiments by Carter give a value for m of approximately 2.6. This expression holds until the plume becomes parallel to shore, at which point the data suggest that the dilution rate increases such that:

$$(18) \quad \theta_{\xi}/\theta_0 = \left\{ \frac{\xi}{nb_0} \right\}^{1/2}$$

where n takes a value such that Equations (17) and (18) give the same dilution for the distance along the axis of the plume $\xi = \xi_m$ corresponding to the point x_m, y_m .

The areas enclosed within specified isotherms in the case of a jet discharge from the shoreline in the presence of a significant longshore current are therefore considerably greater than for the case of zero longshore current speeds. The situation is not significantly improved by moving the jet discharge offshore by means of, for example, a discharge canal, or any other discharge conduit which interferes with the free flow of the ambient current along the shore. If, however, the discharge orifice is moved offshore and the discharge pipelines or conduits are buried below the bottom of the waterway, such that no interference with the ambient flow inshore of the discharge orifice occurs, then, even in the presence of a strong longshore current, dilution rates approaching those for zero longshore currents are likely. The reason for this is that "new" water is made available on the inshore side of the bent jet plume for dilution of the heated effluent by entrainment into the plume. Unfortunately experiments have not been completed which would reveal the degree of improvement in dilution rate as a function of distance offshore at which the discharge orifice is to be located.

III. A Conceptual First-Order Model of the Distribution of Excess Temperature in Thermal Plume

A. General Discussion

On the basis of the discussion given above, we conceive the following sequence of processes which determines the distribution of temperature in a thermal plume resulting from the discharge of heated condenser cooling water from a power plant:

The heated effluent enters the receiving waterbody with a velocity different from the velocity of the ambient currents. Even in the case of existing power stations, where in general no consideration was given in the design to the promotion of rapid mixing of the effluent with the receiving waters, the condenser cooling water flow is usually confined by a bulk-headed canal or a closed conduit at least to the water's edge, and often to some point offshore by jetties. Either by design or as a result of the natural balance of sedimentation and scour at the point of discharge, the condenser cooling water enters the receiving waterway through a defined orifice which gives the heated effluent a definite direction and speed. The direction of discharge is usually approximately normal to the shoreline, and hence the discharge will have an excess of momentum in the offshore direction. In the absence of an ambient longshore current, the excess momentum in the thermal plume formed by the discharge of the condenser cooling water is lost by entrainment; consequently the excess heat in the plume is diluted and the excess temperature is decreased with distance along the plume. In the presence of ambient longshore currents in the receiving waters the thermal plume will initially have zero momentum in a direction parallel to the shoreline in addition to an excess of momentum in the offshore direction. In this case the excess momentum in the offshore direction is lost, and also momentum in the longshore direction is gained as a result of momentum entrainment and work done against the pressure forces. Consequently dilution of the excess heat occurs at a slower than otherwise rate in the case of a discharge into a transverse current.

There will be a region of the thermal plume, which we will call the "near field", within which momentum entrainment is the dominant process controlling the distribution of excess temperature. In this region, in the absence of a longshore current, the excess temperatures along the centerline of the plume vary as the inverse square root of distance. For a bent jet, the excess temperature varies as the inverse fourth root of distance out to about the point at which the plume becomes parallel to shore, beyond which point the excess temperature decreases as the inverse square root of distance.

In the region of the thermal plume where natural mixing processes dominate, a region we will call the "far field", the excess temperature along the centerline varies as the inverse first power of distance.

As a result of surface cooling, heat is being lost from the water to the atmosphere from all areas of the thermal plume, and consequently cooling modifies the distribution of temperature as determined by the processes of momentum entrainment and natural mixing. However, in spite of the fact that the rate of surface heat loss per unit area is greater the larger the excess temperature, in most practical cases dilution so rapidly reduces the area having relatively high excess temperatures that only a small fraction of the rejected heat is lost from areas having excess temperatures exceeding about $1/10$ the temperature rise at the condensers. Most of the rejected heat is lost from areas having relatively small excess temperatures. Cooling thus usually results in only a small modification to the temperature distribution in the thermal plume, for excess temperatures greater than about 10% to 20% of the temperature rise across the condensers.

As a result of the more or less sequential dominance of the three major processes controlling the distribution of excess temperature in the thermal plume, it is possible to proceed in a stepwise manner in using this conceptual model as a basis for predicting the probable distribution of temperature in the receiving waterway. In this procedure, the distribution of excess temperature due to dilution alone (that is, assuming

no surface cooling) is first computed using the relationships given earlier for momentum entrainment and natural mixing. This distribution is then corrected for surface cooling.

The model as discussed above assumes that the receiving water body can be considered essentially as an infinite sink, both from the standpoint of dilution and surface cooling. Where the receiving waterway is not sufficiently large, or the natural processes of water movement and exchange do not supply a sufficient quantity of "new" diluting water to the segment of the waterway adjacent to the plant site, then the model must be appropriately adjusted. The procedure for making such an adjustment is to determine an entrainment excess temperature, which is the excess temperature (corrected for surface cooling) which would exist if the heated effluent were completely mixed into the available rate of supply of diluting water to the segment of the waterway into which the heated effluent is discharged. The diluting water entrained into the thermal plume is then assumed to have this entrainment excess temperature, rather than zero excess temperature, and the computed thermal distribution is appropriately modified.

The relationships given in previous sections allow us to compute the excess temperature distribution along the axis of the thermal plume assuming no heat loss by surface cooling, and provide the basic equations for the surface cooling process. In order to apply the conceptual model outlined above to the prediction of the probable distribution of excess temperature, two additional pieces of information are required.

First, we must know not only the length along the axis of the plume of the area having excess temperatures equal to or greater than some designated value, but also the shape of the isotherm bounding that area. Secondly, the procedure to be employed in using the basic cooling relationships to correct the temperature distribution for surface heat loss must be established.

B. The Shape of the Thermal Plume

As discussed earlier, our present best estimates of the relationships giving the maximum width, b_θ , and the mean areal width $\bar{b}_\theta$, of segments contained within any given isotherm of excess temperature, θ , are

$$(19) \quad \begin{aligned} (a) \quad b_\theta &= \xi_\theta / 4.0 \\ (b) \quad b_\theta &= \xi_\theta / 4.65 \end{aligned}$$

We have used the following guidelines in presenting a schematic depiction of the horizontal distribution of excess temperature in the thermal plume.

- (a) The area contained within a given isotherm θ increases smoothly in width from the discharge orifice, attaining the maximum width, b_θ , at a distance along the plume equal to $0.33 \xi_\theta$
- (b) The area defined by the isotherm θ has a constant width b_θ from a point along the axis of the plume equal to $0.33 \xi_\theta$ to a point equal to $0.75 \xi_\theta$
- (c) The shape of the isotherm θ between the distance $0.75 \xi_\theta$ and the maximum distance ξ_θ is best described as "bullet shaped".

The area, A_θ , contained within an isotherm of excess temperature θ , when shaped as described above, is given by

$$(20) \quad A_\theta = 0.86 b_\theta \xi_\theta = 0.215 \xi_\theta^2$$

It is likely that, as more data becomes available, and as theoretical work on the thermal plume model progresses, these tentative rules will be replaced.

C. The Surface Cooling Correction

After relationships such as Equation (13), or Equations (17) and (18), have been used to determine the distance along the axis of the thermal plume to a specified isotherm of excess temperature, and Equations (19) and (20) have been used to compute the maximum width and the area contained within that isotherm under conditions of zero heat loss across the surface, corrections must be made to this computed distribution of excess temperature to take into account the actual surface cooling. This step in the procedure is based on the following arguments:

The rate of loss of excess heat from an incremental surface area $\Delta A_{\bar{\theta}}$ having a mean excess temperature $\bar{\theta}$, is given by

$$(21) \quad \Delta \Gamma_{\bar{\theta}} = \mu \bar{\theta} \Delta A_{\bar{\theta}}$$

Now designate:

A_{θ_n} = Area of thermal plume inside the isoline of excess temperature θ_n (i.e., the area having excess temperatures $\theta \geq \theta_n$)

Then

$$\Delta A_{\bar{\theta}_n} = A_{\theta_n} - A_{\theta_{n-1}}$$

and

$$\bar{\theta}_n = \frac{\theta_n + \theta_{n-1}}{2}$$

That is, $\Delta A_{\bar{\theta}_n}$ is the incremental area between two adjacent isolines of excess temperature, and $\bar{\theta}_n$ is the mean excess temperature for that incremental area.

The rate of loss of excess heat from the area between two adjacent isolines of excess temperature is then given by

$$(22) \quad \left\{ \Delta \Gamma_{\bar{\theta}_n} \right\}_m = \mu \bar{\theta}_n \left\{ \Delta A_{\bar{\theta}_n} \right\}_m$$

where the subscript $\left\{ \right\}_m$ denotes that these parameters apply to the temperature distribution determined assuming only dilution (mixing) takes place, with no surface cooling. As a result of surface cooling, the areas and heat loss terms will be smaller than those determined for mixing alone. Using the subscript $\left\{ \right\}_{m,c}$ to denote conditions after correction of the excess temperature distribution for surface cooling, then

$$(23) \quad \left\{ \Delta \Gamma_{\bar{\theta}_n} \right\}_{m,c} = \mu_{\bar{\theta}_n} \left\{ \Delta A_{\bar{\theta}_n} \right\}_{m,c}$$

The loss of excess heat by surface cooling can be considered as a decrease in the source strength of the mixing heated effluent. Thus, if Q_h is the rate of heat rejection at the condensers to the condenser cooling water flow, and hence the source strength at the point of discharge for the mixing thermal plume, then the effective source strength applicable to the $\bar{\theta}_n$ isoline of excess temperature, due to surface cooling from the area contained within that isotherm, is given by

$$(24) \quad \left\{ Q_h \right\}_{\bar{\theta}_n} = Q_h \left[1 - \frac{1}{Q_h} \sum_{i=0}^n \left\{ \Delta \Gamma_{\bar{\theta}_i} \right\}_{m,c} \right]$$

where the summation is taken over all incremental areas for isotherms of excess temperature equal to and greater than $\bar{\theta}_n$.

The incremental area $\left\{ \Delta A_{\bar{\theta}_n} \right\}_{m,c}$ is then given by

$$\left\{ \Delta A_{\bar{\theta}_n} \right\}_{m,c} = \left\{ \Delta A_{\bar{\theta}_n} \right\}_m \times \frac{\left\{ Q_h \right\}_{\bar{\theta}_n}}{Q_h}$$

or

$$(25) \quad \left\{ \Delta A_{\bar{\theta}_n} \right\}_{m,c} = \left\{ \Delta A_{\bar{\theta}_n} \right\}_m \times \frac{Q_h - \sum_{i=0}^n \left\{ \Delta \Gamma_{\bar{\theta}_i} \right\}_{m,c}}{Q_h}$$

In order to compute the summation term on the right side of equation (25), however, it is necessary to know $\{\Delta A_{\theta_i}\}_{m,c}$ for all values of i from 1 to n . Combining equations (22), (23), and (25) gives

$$(26) \quad \left\{ \Delta A_{\theta_n} \right\}_{m,c} = \left\{ \Delta A_{\theta_n} \right\}_m \frac{Q_h - \sum_{i=0}^{n-1} \{\Delta \Gamma_{\theta_i}\}_{m,c}}{Q_h + \{\Delta \Gamma_{\theta_n}\}_m}$$

For the innermost isotherm, i.e., for $n = 1$, the summation term on the right side of Equation (26) is zero, and the corrected area $\{\Delta A_{\theta_n}\}_{m,c}$ is immediately available from Equation (26) using the $\{\Delta \Gamma_{\theta_n}\}_m$ computed from Equation (22). Then, computing $\{\Delta \Gamma_{\theta_n}\}_{m,c}$ from Equation (23), the corrected area between the next lower adjacent isolines of excess temperature can be obtained. This procedure is repeated until all incremental areas and heat loss terms have been corrected.

The corrected areas, $\{A_{\theta_n}\}_{m,c}$ having excess temperatures equal to or greater than θ_n , for the combined effect of both mixing and cooling, are then obtained by simply summing the corrected incremental areas

$\{\Delta A_{\theta_i}\}_{m,c}$ from $i = 1$ to $i = n$.

IV. Example computations Using the Above-Described Model

The relative importance of the several processes which control the distribution of temperature in the thermal plume resulting from the discharge of condenser cooling water to a natural water body can be shown by comparing the results of several illustrative example computations using the model described above.

The following five example computations serve to provide a comparison between the distributions of excess temperature in the thermal plumes resulting from the discharge of condenser cooling water via discharge structures designed

either to minimize dilution of the heated effluent or to promote such dilution. In each of the five cases described below it is assumed that a steam-electric generating plant rejecting 6.8×10^9 BTU/hr of excess heat at the condenser discharges the heated condenser cooling water into the surface layers of the water body. This rate of heat rejection corresponds to a nominal 1000 MWE nuclear power plant or to a nominal 1700 MWE fossil fuel power plant, at present efficiency levels. It is further assumed that the temperature rise of the condenser cooling water flow at the condensers for the first four cases is 20°F, and hence the required volume rate of flow of condenser cooling water is 1520 ft³/sec. For Case V it is assumed that the temperature rise at the condensers is 10°F, and hence the required volume rate of flow of condenser cooling water is 3040 ft³ · sec⁻¹. In all cases it is also assumed that the heated effluent is discharged into a waterway having no longshore currents, or that the discharge structure is located far enough offshore so that free passage of dilution water can occur on the inshore side of the bent thermal plume even in the presence of a longshore current.

Case I: Discharge structure designed so that no dilution (mixing) of the heated effluent with the receiving waters of the Lake occurs. Hence the decrease in excess temperature occurs as a result of surface cooling only. It should be recognized that it is in fact impossible to discharge the condenser cooling water flow into the Lake without some mixing between the heated effluent and the cooler receiving waters. However, it is of some interest to consider this case for purpose of comparison.

Case II: Discharge structure designed to minimize dilution of the heated effluent by the cooler receiving waters of the Lake. Dilution of the thermal plume by both momentum entrainment and turbulent diffusion is enhanced by introducing the heated effluent as a high velocity jet. Hence in order to minimize dilution of the thermal plume the discharge structure should be designed to produce as low a discharge velocity of the condenser cooling water flow as

possible. The conditions assumed for this case produce what we consider to be a practical minimum of mixing of the thermal plume with the receiving waters. Specifically, we have assumed a discharge orifice 500 ft. wide and 10 ft. deep, discharging into the surface layers of the Lake at a point where the water depth is at least 10 feet. This discharge orifice would result in a discharge velocity of the heated effluent of 0.30 ft/sec.

Case III: Discharge structure designed to produce a rate of dilution of the thermal plume by the receiving waters "typical" of many existing steam-generating electric plants. Specifically, we have assumed a discharge orifice 50 ft. wide and 10 ft. deep, discharging into the surface layers of the Lake at a point where the depth is at least 10 feet. This discharge orifice would result in a discharge velocity of the heated effluent of 3.0 ft/sec.

Case IV: Discharge structure designed to promote rapid dilution of the thermal plume by the receiving waters. The design criteria for the discharge structure assumed for this case do not represent the most optimum design criteria for producing the maximum dilution possible. However, they do represent criteria which can be met with reasonable engineering effort and serve to indicate the type of thermal distribution which can be attained by using the basic design principles which have been applied to the discharge structures of a number of power plants now under construction or in the planning stage. These include the Surry Nuclear Power Station on the James River estuary, the Morgantown Power Plant on the Potomac River estuary, the Calvert Cliffs Nuclear Power Plant on the Chesapeake Bay, the Pilgrim Nuclear Power Station on Cape Cod Bay, and the Zion Nuclear Power Station on Lake Michigan. Specifically, we have assumed a discharge orifice 15 feet wide and 10 feet deep, discharging into the surface layers of the Lake at a point where

the water depth is at least 10 feet. This discharge orifice would result in a discharge velocity of the heated effluent of 10.1 ft/sec.

Case V: Same as Case IV, except that with a temperature rise at the condensers of 10°F instead of 20°F, with the consequent condenser cooling water flow of 3040 cfs instead of 1520 cfs, the width of the orifice must be 30 ft. rather than 15 ft. for the discharge velocity to remain at 10.1 ft · sec⁻¹.

Table 3 gives the results of the computations for the five cases described above, in terms of the area, in acres, contained within the specific isotherms of excess temperature.

The computational model also gives the length, width, and general shape of these areas contained within the specified isotherms, and hence the horizontal distribution of excess temperature in the surface layers of the water body can be given in the form of contours of excess temperature on a schematic plan view of the water body adjacent to the point of discharge, as shown in Figure 1. Since Case I does not represent a practically attainable situation, this case is not included in Figure 1. Also note that Case V is not shown in this figure. The 1°F and 2°F excess temperature isotherms for Case V would very nearly coincide with those of Case IV, while the 4° and 6°F isotherms of excess temperature for Case V would extend about one-half as far as for Case IV, and enclose a surface area about one-quarter as large as for Case IV. There would of course be no excess temperatures greater than 10°F for Case V.

Table 3

The Area, in Acres, Contained Within Specified Isotherms of Excess Temperature in the Thermal Plume Resulting from the Discharge of the Condenser Cooling Water Flow from a Steam-Electric Generating Plant, Rejecting 6.8×10^9 BTU/hr of Excess Heat at the Condensers, for Five Different Discharge Structure Designs as Described in the Text. (This Rate of Heat Rejection Corresponds to a Nominal 1000 MWE Nuclear Power Plant or to a Nominal 1700 MWE Fossil Fuel Plant at Present Efficiency Levels).

	Case I	Case II	Case III	Case IV	Case V
$\theta_o(^{\circ}\text{F})$	20	20	20	20	10
$Q_c(\text{ft}^3\text{s}^{-1})$	1520	1520	1520	1520	3040
$b_o(\text{ft})$	(cooling	500	50	15	30
$u_j(\text{ft s}^{-1})$	only)	0.30	3.0	10.1	10.1

$\theta^{\circ}\text{F}$	Area, in Acres, for:				
14	3.78×10^2	1.66×10^2	1.9	0.2	----
10	7.71×10^2	5.42×10^2	7.1	0.6	----
5	1.66×10^3	2.57×10^3	1.11×10^2	10	2.6
3	2.36×10^3	3.19×10^3	4.65×10^2	44	20
2	2.94×10^3	3.26×10^3	1.00×10^3	99	1.0×10^2
1	3.90×10^3	3.29×10^3	3.47×10^3	3.91×10^2	4.0×10^2

Fig. 1. Horizontal distribution of excess temperature at the surface in °F for Cases II, III, and IV (see text).

V. Temperature-Time of Exposure Relationships for Organisms Entrained into the Thermal Plume

A number of biologists who have studied the effects of temperature increases on aquatic and marine biota in the laboratory have been surprised to find, upon extending their studies to actual thermal discharges in natural water bodies, that the conclusions they had arrived at from their laboratory studies were not applicable to the real world. The major reason for this apparent contradiction is that most laboratory studies have involved exposures of organisms to temperature increases for relatively long time periods. These investigators failed to realize that the biologically important relationship to be investigated is the temperature-time exposure history. The discharge structures of many existing power plants provide for sufficiently rapid reduction of the temperature in the thermal plume due to dilution so that the laboratory data on mortality and other detrimental effects on organisms simply do not apply.

It is possible to develop relationships which give the maximum time of exposure to any specified temperature rise for organisms entrained into the thermal plume, for any specified set of discharge structure design criteria. For the case of surface cooling only (Case I), the time interval, τ_{θ} to which organisms entrained in the thermal plume would be exposed to excess temperatures equal to or greater than θ , is given by

$$(27) \quad \tau_{\theta} = \frac{\rho C_p D_h}{\mu} \ln \frac{\theta_o}{\theta}$$

Note that this time interval depends on the depth to which the excess heat is distributed. For our calculations we have taken a value of D_h of 5 feet and a value of μ , the surface cooling coefficient, corresponding to a wind speed of 10 mph.

For the case of both dilution and cooling the exposure time τ_{θ} is given by:

$$(28) \quad \tau_{\theta} = \frac{0.272 \xi_{\theta}^{3/2}}{(u_j)_o b_o^{1/2}} \quad \text{for } \theta \geq 0.2\theta_o$$

and

$$(29) \quad \tau_{\theta} = \tau_{0.2\theta} + \frac{\{\xi_{\theta}^2 - \xi_{0.2\theta}^2\}}{60 b_o (u_j)_o} \quad \text{for } \theta \leq 0.2\theta_o$$

where $(u_j)_o$ is the speed of the jet discharge at the orifice; b_o is the width of the discharge orifice; ξ_{θ} is the distance along the axis of the plume to the excess temperature isotherm θ , as corrected for surface cooling; $\xi_{0.2\theta}$ is the distance along the axis of the plume to the excess temperature isotherm $\theta = 0.2 \theta_o$ uncorrected for surface cooling; and $\tau_{0.2\theta}$ is the time computed for the distance $\xi_{0.2\theta}$ using Equation (28).

These equations have been used to compute the time-temperature relationships for each of the five cases of discharge design described earlier in this paper. The results of these computations are shown in Table 4, in which the maximum time during which an organism entrained into the thermal plume would be subjected to excess temperatures equal to or greater than the listed values is given.

Note that the times of exposure given in Table 4 are measured from the time of discharge of the heated effluent into the surface layers of the water body. The time of transit of the condenser cooling water flow from the condensers to the point of discharge must be added to the times given in Table 4 in order to obtain the total maximum time of exposure as a function of excess temperature for organisms which are entrained into the condenser cooling water intake flow and carried with the flow through the plant. For several recently designed power stations the time of transit of the condenser cooling water from the condensers to the point of discharge will be on the order of 2 to 3 minutes. For some older plants, particularly those which discharge the heated effluent via long discharge canals which generally afford negligible cooling, the time of exposure of organisms carried through

Table 4

Maximum Time of Exposure vs. Excess Temperature for Organisms Entrained Into the Thermal Plume Resulting from the Discharge of the Condenser Cooling Water Flow from a 1000 MWE (Nuclear) or a 1700 MWE (Fossil Fueled) Power Plant, for the Five Cases of Discharge Structure Design as Described in the Text.

	Case I	Case II	Case III	Case IV	Case V
$\theta_o (^{\circ}\text{F})$	20	20	20	20	10
$Q_c (\text{ft}^3 \text{s}^{-1})$	1520	1520	1520	1520	3040
$b_o (\text{ft})$	(cooling	500	50	15	30
$u_j (\text{ft s}^{-1})$	only)	0.30	3.0	10.1	10.1

$\theta^{\circ}\text{F}$	Time of Exposure For:				
14	17 hr	5 hr	3.0 min	17.0 sec	----
10	32 hr	12 hr	9.0 min	47.0 sec	----
5	64 hr	38 hr	1.2 hr	6.0 min	1.6 min
3	88 hr	41 hr	3.4 hr	19.0 min	7.3 min
2	107 hr	42 hr	7.0 hr	40.0 min	25.0 min
1	139 hr	43 hr	22.0 hr	2.4 hr	1.3 hr

the plant in the condenser cooling water flow to the maximum temperature rise is on the order of 2 hours.

VI. Biological Considerations

The effects of the use of water drawn from a natural water body for cooling of the condensers in a power station, and subsequent discharge of the heated effluent back into the waterway will depend not only on the design features of the cooling water system, but also on the peculiar ecology of the water body. Consequently many biologists hesitate to make general statements which might assist in establishing design and siting criteria, and instead insist that each case must be the subject of individual, prolonged study.

We believe, however, that enough is now known to provide a basis for establishing design and siting criteria in order that once-through cooling systems would produce a minimum, and hopefully negligible, detrimental effect on the aquatic biota in the waterway adjacent to the plant site.

There is adequate evidence to show that phytoplankton, zooplankton, and fish eggs and larvae drawn through the condensers with the condenser cooling water flow are subjected to a fractional kill. The fraction of any given species killed during such passage has been observed to be highly variable in time and also varies considerably from one species to another. Most studies of the mortalities of organisms entrained into the condenser cooling water flow and passed through the condensers have failed to distinguish between three possible causes for such kills; i. e.: (1) mechanical damage; (2) biocide effects (usually chlorine); and (3) thermal shock. Recent evidence (16) suggests that most of the effects which have been observed on phytoplankton probably result from chlorination. Since chlorine is introduced into the condenser cooling water flow in order to reduce or eliminate fouling of the condenser tubes by sessile organisms, it is not surprising that this non-specific biocide exacts a toll on the entrained organism. We believe that a major portion of the mortality of the zooplankton

and fish larvae which have been observed in the condenser cooling water flow is caused by the use of chlorine or some other biocide.

Some limited kills of organisms passing through the cooling water system have been observed when the plant was shut down except for the pumps required to circulate the cooling water. Thus some mechanical damage can occur to certain zooplankton in the absence of both biocide and thermal effects.

The case for actual thermal kills of organisms passing through the condensers is based primarily on laboratory studies, since field observations have not adequately separated the thermal effects from other possibly detrimental features of the cooling water system. Most of these laboratory studies have involved exposures of organisms to the maximum temperature rise for periods of from several hours to several days. Only recently has there been any effort to study, in the laboratory, the effects of subjecting aquatic organisms to a temperature-time history more nearly that which occurs in actual operating of power plants or which will occur in some of the recently designed stations. For example, early laboratory studies of thermal effects on organisms in the Patuxent River estuary made use of 24-hour exposures. Later, the work was extended to cover exposure periods of from one to three hours, which is in the range of exposure of organisms which are entrained into the condenser cooling water flow of the Chalk Point Power Plant on the Patuxent. Only very recently have efforts been made to study the effects of exposing estuarine organisms to a time-temperature history similar to those which will prevail in the Calvert Cliffs Nuclear Power Plant on the Chesapeake Bay. In this plant, time of passage of the condenser cooling water from the condensers to the point of discharge will be on the order of 3 minutes. Because of the high velocity discharge, subsequent dilution and cooling will reduce the excess temperature from its initial value of 10°F to less than 5°F in about 1.5 minutes, and to 3°F in less than 8 minutes. Oyster larvae, a particularly important and sensitive transient member of the

zooplankton in estuaries, has been shown to suffer significant mortalities when exposed to a ten-degree temperature rise from a base temperature of about 80°F for periods of two hours or more. When the larvae were exposed to a time-temperature history similar to that described for the Calvert Cliffs Plant, insignificant mortality occurred.

Most detrimental effects which might occur in the waterway itself as a result of the thermal plume will also depend on the time-temperature history of organisms exposed to the heated effluent. Most highly mobile organisms have been shown to be able to detect an adverse temperature gradient, and to avoid the heated water in the plume if the temperatures in the plume are above the optimum range. The less mobile members of the plankton may be entrained into the plume along with the diluting water. Hence the detrimental effects on such entrained organisms is dependent on the time-temperature history as described in the previous section of this paper.

Sessile bottom organisms cannot escape undesirably high temperatures in a plume which impinges on the bottom area where they reside, nor would they be moved through the plume by the induced flow to regions of lower temperature. Hence these organisms could be exposed to the elevated temperature in the thermal plume for long periods of time, at least to the extent that the current regime was steady.

There has been considerable concern about the development of a so-called thermal block which would prohibit migrating fish from traveling up or down the waterway enroute to or from important spawning or nursery areas. Such concern has been reflected in the thermal standards in several states, which require that some fraction of the cross-section of a river or estuary be free of any temperature rise above ambient. We consider such regulations to be impossible to enforce, because of the considerable difficulty in establishing the base or ambient temperature which would prevail in the absence of any thermal discharge, and also to be practically impossible to accomplish, since at some perhaps rather

great distance from the plant some excess heat, however, small, will extend across the waterway. This goal also implies a design criteria which is incompatible with other, more important goals for protection of the environment from detrimental effects of power plant operation. We believe that the possibility of a thermal block can be eliminated if the areas enclosed by the higher excess temperature isotherms are kept small relative to the surface area of the segment of the waterway into which the discharge of heated effluent is made, and the spacial gradients in temperature (that is, the rate at which temperature changes with distance) are kept as small as possible, except for a very small area around the point of discharge.

The discussion so far has dealt with the biological consequences of passage of organisms through the condenser cooling water system and exposure of organisms to the thermal plume in the waterway. A considerable fraction of all the reported kills of aquatic organisms which have been judged to result from power plant operation has been related to the condenser cooling water intake structure rather than any process involving the discharge. The impinging of large numbers of fish upon the intake screens has occurred, and appears to be particularly severe at some plants located on estuaries. Most power plants have been built with little thought given to the design of the intake structure so as to guide fish away from the often trap-like feature of the intake.

VII. Desirable Features of a Once-Through Cooling Water System From An Environmental Standpoint

On the basis of the discussion given above, the following conclusions are reached as to the desirable features of a once-through cooling system from the standpoint of minimizing detrimental effects on the aquatic environment to the end that such effects will be negligible:

1. Organisms carried through the condensers with the condenser cooling water flow should be exposed to excess temperatures at or close to the temperature rise across the condensers for as short a time period as practicable.
2. Organisms carried into the thermal plume with the dilution water as a result of momentum entrainment should experience as rapid a decrease in excess temperature as practicable.
3. Organisms entrained into the condenser cooling water flow at the intake, or into the thermal plume after discharge should be subjected to exposure to any non-specific biocide to the minimum extent practicable.
4. The area of the bottom of the waterway which may be exposed to excess temperatures in the thermal plume of a magnitude sufficient to cause detrimental effects to benthic organisms (not withstanding the exposure time required) should be kept as small as practicable.
5. Except for an area immediately around the point of discharge, having linear dimensions small compared to the distance across the waterway, the spacial gradients in temperature should be kept as small as practicable.
6. Mobile organisms (i.e., finfish, crabs etc.) should be subject to as low an intake velocity as practicable, and should be diverted from the fact of the intake structure.

VIII. Design and siting Criteria

On the basis of the analysis of (1) the distribution of excess temperature in the thermal plume in a natural water body resulting from the discharge of condenser cooling water from a steam-electric generating plant, and (2) the time-temperature history of organisms entrained into

the condenser cooling water at the intake or into the thermal plume after discharge, the following criteria for the design of once-through cooling systems and the siting of power plants using such systems are indicated in order to provide maximum environmental protection:

1. The plant should be located on water bodies having large potential diluting capacity. Specifically, the volume rate of supply of new dilution water made available to the segment of the waterway into which the heated discharge is to be made should be at least five times the volume rate of flow of the condenser cooling water supply, assuming that the temperature rise at the condenser remains in the range now common for once-through systems. A large potential diluting capacity, when coupled with design features to be listed below, provides for (a) a low background excess temperature of the diluting water entrained into the thermal plume, thus minimizing the areas and static volumes of water having relatively large excess temperatures, and (b) limiting the fraction of the zooplankton population of the waterway which is carried through the condenser cooling water system, where a maximum possibility of damage to the organisms exists, even if only from mechanical shock.
2. The water body should have depths at a reasonable distance from the shoreline at the plant site which are greater than the probable vertical thickness of the thermal plume in the region of high excess temperatures. The purpose of this siting requirement is to limit the bottom area where benthic organisms might be subjected to prolonged periods of exposure to detrimental excess temperatures. Furthermore, vertical entrainment and mixing through the bottom of the thermal plume, even though this mechanism is not the major diluting process, is enhanced if the water depths are large compared to the vertical thickness of the thermal plume. If the water depths are sufficiently large, the rate of dilution of the heated effluent can be

enhanced by discharging the effluent as a high speed horizontal jet at the bottom. Due to its buoyancy, the plume will rise towards the surface, and entrainment will occur both as a result of the excess horizontal momentum, and as a result of the vertical movement of the plume through the water column.

3. The discharge structure should be designed to produce as high a velocity of discharge as is practicable within the limits set by head loss and possibly by navigation problems. The purpose of this requirement is to provide for most rapid dilution of the heated effluent, for the smallest area and static volume of water having excess temperatures greater than about 10% of the temperature rise at the condensers, and for the most favorable time-temperature history for organisms entrained into the condenser cooling water flow or into the thermal plume. From the standpoint of possible navigation hazards for small boat traffic, a submerged jet discharge may have some advantage over a surface discharge. Note that the combination of this design feature with the siting requirement for large potential diluting capacity implies that such diluting capacity need not be provided by a steady flow into the subject segment of the water body from adjacent segments. The momentum jet entrainment of water into a high speed jet will set up large scale circulations which will bring diluting water from adjacent segments of the waterway. Thus a large lake, which may not have a circulation pattern which provides a natural supply of new diluting water to the segment of the lake into which the discharge is to be made, may still satisfy the requirements of (1) above if the discharge structure is properly designed.

4. In water bodies in which tidal processes, wind stresses, or other natural forces result in longshore current flowing transverse to the direction of the jet discharge with speeds of one-tenth or more

of the speed of the heated effluent at the discharge orifice, the rate of dilution of a thermal plume originating at a shoreline or nearshore surface discharge structure will be significantly reduced compared to the case of zero longshore current speeds. This reduction in the dilution rate can be avoided by locating the discharge structure a sufficient distance offshore so that dilution water required for the inshore side of the plume can pass freely between the discharge orifice and the shore. Note that to accomplish this design criteria, the pipes or conduits carrying the heated discharge must be buried beneath the bottom of the waterway between the shoreline and the point of discharge.

5. In order to minimize the time period over which organisms carried through the plant in the condenser cooling water flow are exposed to the maximum temperature rise, the time of passage of the cooling water flow from the condensers to the point of discharge should be as small as practically possible. Long discharge canals with flow velocities of only a few feet per second are definitely undesirable. Despite some opinion to the contrary, such canals afford insignificant cooling of the heated effluent. Even though such canals often provide favorable sport fishing during the cooler seasons, this advantage is outweighed by the long time period of exposure of entrained organisms. Also such canals provide a type of attractive trap where kills of finfish and other relatively large mobile species have occurred due to accidental over-application of biocide, since the canal does not afford any dilution of the biocide.

6. Fouling of the condenser tubes should be controlled by either mechanical methods or by the use of biocides or surface repellants specific to the fouling species.

7. In order to minimize the possibility of impinging large numbers of fish or other mobile species on the intake screens, the intake

velocity in front of the screens should be kept low (on the order of $0.5 \text{ ft} \cdot \text{sec}^{-1}$), and some method of diverting such forms away from the front of the intake screen should be provided. Though not yet fully developed, horizontal traveling screens appear to offer considerable promise for installations on rivers and tidal estuaries for this purpose.

It should be made clear that conditions peculiar to a given waterway or location may alter one or more of the above general design and siting criteria.

A comparison of the data given for Case IV and Case V in Table 3 and Table 4 suggests that, for a given size plant and a given speed of jet discharge of the heated effluent, there is some apparent advantage to a lower temperature rise at the condensers, and a consequent higher volume rate of flow of condenser cooling water. Table 3 shows that for excess temperatures greater than 2°F Case V (with a 10°F temperature rise at the condensers and a cooling water flow of 3040 cfs) has smaller areas in the thermal plume than does Case IV, for which the temperature rise at the condensers was taken to be 20°F , and the cooling water flow was consequently 1520 cfs. Table 4 shows that entrained organisms would be exposed to any excess temperature greater than 1°F for a smaller time interval for Case V than for Case IV.

These apparent advantages may not be real, since the volume rate of flow of cooling water, and hence the number of entrained organisms drawn through the condensers per unit time, would be twice as great in Case V than in Case IV. Since there may be mechanically induced damage to some fraction of the organisms passing through the cooling water system, and since the areas enclosed by excess temperature isotherms having possibly detrimental effects, and the time of exposure of entrained organisms to such temperatures, are quite small for Case IV, even if not as small as for Case V, it may be better to use the smaller amount of water and have the larger temperature rise at the

condensers. At present there is not adequate biological information as to which of these two cases, (i. e., Case IV or Case V), would provide the least potential damage to entrained organisms. From a practical standpoint, the engineering problems associated with construction of an offshore discharge structure for a high speed discharge of the heated effluent are much less for the smaller volume rate of flow of cooling water. Because of the smaller sized pipe or conduit required for transport of the condenser cooling water flow, the discharge orifice could be placed farther offshore for Case IV than for Case V, thus providing greater assurance of rapid reduction in excess temperature, both in space and in time.

IX. Some Comments on the Relative Importance of Heat Rejected to the Environment in the Production of Electric Power

The present annual consumption of thermal energy in the United States amounts to about 6.25×10^{16} BTU. This corresponds to an average consumption rate of 7.13×10^{12} BTU $\cdot$ hr⁻¹, or 2.09×10^9 KWT. For a current population of 204 million, this represents an average per capita rate of consumption of thermal energy of 10.25 KWT.

Table 5 shows how this average per capita rate of consumption of thermal energy is divided between various segments of society; i. e.: domestic (household), commercial, transport, and industry; and between various use categories; i. e.: space heating, other heating, electricity, motive use, and non-energy use. Note that the rate of consumption of thermal energy for the production of electricity averages 2.10 KWT per capita, and this amount is very nearly equaled by the per capita consumption of thermal energy in the household (2.05 KWT) and for transport (2.05 KWT).

At the present average efficiency of conversion of thermal energy to electricity, the per capita production of electric power is, averaged over the year, about 0.84 KWE out of the 2.10 KWT per capita rate of

consumption of thermal energy for the production of this electric power. The remainder, or 1.26 KWT per capita, represents that part of the 10.25 KWT per capita rate of consumption of thermal energy which is rejected to the environment in the production of electric power.

To put this matter another way, of the present average rate of consumption of thermal energy in the United States of $7.13 \times 10^{12} \text{ BTU} \cdot \text{hr}^{-1}$, about 12.3%, or $8.76 \times 10^{11} \text{ BTU} \cdot \text{hr}^{-1}$ is the average rate of rejection of thermal energy (heat) to the environment as a result of the production of electricity. However, the full $7.13 \times 10^{12} \text{ BTU} \cdot \text{hr}^{-1}$, or 10.25 KWT per capita, must ultimately be lost to the environment, by such routes as the heat exchanged to the atmosphere in the cooling system of an automobile, or the friction of tires on a highway, or the heat lost to the air in the process of cooking food, and by many other such processes.

This man-made heat source represents an additional source term in the heat budget of the near surface layers of the atmosphere and the surface of the earth. Consider now the natural input terms in this heat budget equation.

The average input of short wave solar radiation which is absorbed at the earth's surface of the United States, including the surface of natural water bodies, is about $0.18 \text{ KW} \cdot \text{m}^{-2}$, or, since the surface area of the United States is $7.83 \times 10^{12} \text{ m}^2$, the total rate of input of heat to the surface from this source is $1.41 \times 10^{12} \text{ KWT}$. The average input of long wave radiation from the atmosphere to the surface of the United States is $0.36 \text{ KW} \cdot \text{m}^{-2}$, or the total rate of input of heat to the surface from this source is $2.82 \times 10^{12} \text{ KWT}$. The sum of these two natural inputs is then $4.23 \times 10^{12} \text{ KWT}$, or $2.07 \times 10^6 \text{ KWT}$ per capita. Comparing this figure with the 10.25 KWT per capita representing the rate of consumption of man-made energy in the U.S., and the 1.26 KWT representing the per capita rejection of heat to the environment in the production of electricity, reveals the relative small impact man-made sources can have on the overall heat budget.

Table 5

Matrix Indicating the Manner in Which the Per Capita Rate of Consumption of Thermal Energy of 10.25 KWT is Divided Between Various Segments of Society (i.e., Domestic, Commercial, etc.) and also Between Various Use Categories (Space Heating, Electricity, etc.)

Use Category	Segment					Totals
	Domestic	Commercial	Transport	Industry	Other	
Space Heating	1.20	0.15	0.05	0.15	0.10	1.65
Other Heat	0.35	0.40	----	2.00	0.60	3.35
Electricity	0.50	0.30	----	1.00	0.30	2.10
Motive Use	----	----	2.00	----	----	2.00
Non-Energy Use	----	----	----	0.45	0.70	1.15
Totals	2.05	0.85	2.05	3.60	1.70	10.25

Even so, the man-made sources of heat must contribute to some, however slight, increase in temperature of the lower layers of the atmosphere, and hence increase the long wave radiation input term to the near surface heat budget. In response, in order to maintain radiative balance, the heat loss terms from the earth's surface, including water surface, must increase. This can only occur if the temperature of the surface of the earth, including water surfaces, increases, however slight such increase might be. We suggest that the overall average increase in the temperature of the surface waters of the United States, due to man's activity, will not differ significantly whether or not 12.3% of the man-made source term to the near surface heat budget is passed directly to the atmosphere via cooling towers, or indirectly by using the surface waters of this country as transient receivers of the rejected heat from electric power production.

The major problem with respect to using surface waters for condenser cooling in steam-electric generating stations is then not the overall average increase in temperature of these waters, but rather that such inputs of rejected heat tend to be localized; hence local areas of undesirable high water temperatures might occur and perhaps in some cases already have occurred. The use of proper siting and design criteria for once-through cooling systems can provide for the rapid dilution and spread of the rejected heat over large areas at low concentrations, and hence effectively limit the local areas having biologically significant temperature increases.

As pointed out above the overall impact of the rejection of the waste heat produced in the generation of electric power in the United States on the heat budget of the near surface layers of the atmosphere must be of virtually no consequence, even if passed directly to the atmosphere via natural water bodies. Possible undesirable effects of this man-made heat source result from the fact that the input to the atmosphere may be highly localized, thus possibly influencing the nearby micro-climatology. Without passing judgment on how significant such local effects might be, it is worth noting that the consequences of direct passage of heat to the atmosphere via cooling towers does involve a highly localized source, as compared to the much

more dispersed input when the rejected heat is discharged via well designed discharged structures into a natural water body of adequate dilution capacity.

For example, an evaporative cooling tower for a nominal 1000 MWE nuclear power plant, rejecting $6.8 \times 10^9 \text{ BTU} \cdot \text{hr}^{-1}$ to the cooling water flow at the condensers, would release approximately $6.0 \times 10^6 \text{ lbs} \cdot \text{hr}$ of water vapor to the atmosphere. If it is assumed that this vapor is mixed into a volume of flowing air 500 feet wide, 200 feet high, moving at 10 mph, the resulting increase in relative humidity of this air plume, for an ambient temperature of 70°F is about 82%. On the other hand, the added evaporative loss of water from a natural water body which receives this same rate of input of rejected heat via a discharge system as described for Case IV above, would be about $3.8 \times 10^6 \text{ lbs} \cdot \text{hr}$. Assuming that this added water vapor is confined to a layer 30 feet thick over the water surface, in a wind of 10 mph moving across the thermal plume, the resulting increase in relative humidity of the air over the water would nowhere exceed 3%, again for an ambient temperature of 70°F . We would conclude that there is a reasonable possibility of an increase in the incidence of localized fog in the downwind direction from the evaporative cooling tower. There is on the other hand little probability that the incidence of fog would be increased as the result of the discharge of the excess heat through a properly designed once-through cooling system into a natural waterway of adequate dilution capacity.

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