Investigations of Sparging as a Method for Promoting Cooling Pond Heat Transfer[1]

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ABSTRACT

Destruction of the thin subsurface thermal boundary layer at an air-water interface can be accomplished by relatively low rates of aeration and can result in substantially improved thermal performance when water temperatures are high. The heating and saturating of rising air bubbles can also provide a significant improvement in overall thermal performance when water temperatures and aeration rates are sufficiently great. At 80°C, improvements of ~20% appear possible with average aeration rates <1 mm s^{-1}.

1. Introduction

Heat loss from the water surface of lakes and ponds is influenced considerably by a thin, pseudo-laminar sublayer immediately beneath the surface. Measurements made by infrared thermometry, for example, indicate that surface temperatures can be more than 1°C cooler than would be measured with an immersed thermometer, even when it is carefully located in the upper few millimeters of water. For open ocean, this effect has sometimes been suggested as a reason for the appearance of different bulk transfer coefficients for sensible heat and water vapor; this discrepancy was not found at a cooling pond where surface temperature was measured directly (see Hicks *et al.*, 1977). In fact, it is the case of industrial cooling ponds which represents the main incentive for the present consideration of this thin thermal film, since it is conceptually possible to disrupt the layer artificially in order to promote heat exchange with the air. A number of methods for achieving this goal seem feasible, including mechanical mixing, agitation by submerged sonic devices and aeration (or sparging).

Early experimental studies[2] indicated that exceedingly low rates of aeration produced detectable improvements in the efficiency of thermal coupling between a water surface and the air. In these early studies, the aeration process was selected for scrutiny because of the ready availability of suitable equipment. For example, sparging by submerged perforated air hoses is a common method for promoting biological activity in sewage treatment plants. At that time, little was known about the behavior of the thermal boundary layer in other than natural conditions; accordingly, interpretation of the results obtained presented considerable difficulty. Nevertheless, it was concluded that aeration might offer an economically attractive alternative to "spray" cooling as a supplement to existing ponds and canals. The matter was somewhat clouded by the lack of a physically based model capable of addressing situations of high water temperatures and of very low sparging rates. The present intent is to reexamine the matter, starting with the models of thermal skin behavior developed in recent investigations at the cooling lake of the Commonwealth Edison Dresden plant near Morris, Illinois, as reported by Wesely (1979).

2. Thermal skin theory

Many authors have investigated the problem of how to parameterize the subsurface, semi-laminar layer that is usually found at sea. For example, expressions for the thickness of the layer and for the temperature drop across it have been presented by Saunders (1967) and Deacon (1977). Models of this kind tend to agree on the important parameters involved, but there are a number of subtleties when it comes to the matter of how to extend the oceanic results to the case of confined and artificially heated water bodies. The role of turbulence in the water is obviously critical, whether or not it results from thermal activity or wave-induced mixing (see Liu and Businger, 1975). However, the work reported by Wesely (1979) refers directly to the problem of interest here, namely that of small, relatively

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[1] Work performed under the auspices of the U.S. Department of Energy.

[2] Hicks, B. B., C. M. Sheih and G. A. Zerbe, 1974; On the heat transfer properties of an aerated water surface. Argonne National Laboratory Radiological and Environment Research Division Annual Report, ANL-75-3 Part IV, January–December 1974, 54–61.

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shallow bodies of strongly heated water. The temperature drop across the thermal skin as found by Wesely is given by

$$\Delta T_s = C\gamma^{2/3}\kappa^{1/3}QK^{-1}(-\tau/\rho_w)^{-1/2}, \qquad (1)$$

in which the various quantities[3] can be combined in dimensionless groups (such as the Prandtl number, $\mathrm{Pr} = \gamma/\kappa$) similar to those favored by investigators of the oceanic case. Eq. (1) is quite similar to Deacon's result, although there is some difference in the evaluations of the empirical constant C; Wesely obtains $C = 11.7$, whereas the results of Deacon's Fig. 1 indicate $C \approx 13.7$.

The heat transfer rate through the thermal sublayer can be related to the more familiar atmospheric components via

$$Q = H + L_w E + R\uparrow - R\downarrow. \qquad (2)$$

For the present purposes, we can neglect shortwave radiation components (such as the insolation itself) since none of these components is directly influenced by changes in T_s and since it is the effects of such changes that we desire to investigate. Moreover, relatively little of the incoming shortwave radiation will be absorbed by the topmost few millimeters of the water that constitute the thermal diffusive sublayer. It is also apparent that we need not be too concerned about the incoming longwave radiation component $R\downarrow$, since it will also be largely insensitive to T_s.

The sensible and latent heat fluxes in Eq. (2) can be conveniently expressed in terms of bulk aerodynamic approximations in which the empirical exchange coefficients (the Stanton number and the Dalton coefficient, respectively) are essentially the same. Thus we can write

$$Q = \rho_a c_p \bar{u} D_H[T_s - T_a + (L_w/c_p)(q_s - q_a)] + \epsilon\sigma T_s^4 - R\downarrow. \qquad (3)$$

The rate of change of Q with T_s can then be obtained by differentiation as

$$\partial Q/\partial T_s = \rho_a c_p \bar{u} D_H(1 + s/\gamma_a) + 4\epsilon\sigma T_s^3, \qquad (4)$$

where the property s is evaluated at temperature T_s.

A large number of simplifications are included in Eqs. (3) and (4). It is assumed that shortwave radiation components are unaffected by surface temperature changes and that the incoming infrared radiation is also insensitive to T_s. These assumptions are probably defensible when fog is not generated at the surface. The case in which fog occurs will clearly be substantially different. It is also obvious that the transfer coefficient D_H must be a function of T_s, since it is well known that changes in atmospheric stability will strongly affect all coefficients of this kind and the stability index will be determined by the air-water temperature difference. For the present, variations in D_H will not be considered. Later, it will become apparent that the omission is not a matter of great importance.

The magnitude of the enhancement of heat exchange with the atmosphere when the thermal sublayer is mechanically destroyed can now be estimated. The change in Q corresponding to a surface temperature variation ΔT_s can be approximated directly from Eq. (4) as $\Delta T_s(\partial Q/\partial T_s)$. By substituting for ΔT_s with Eq. (1) we can derive an estimate of the proportional change in Q as

$$(\Delta Q/Q) \approx (\partial Q/\partial T_s)(B/u_*), \qquad (5)$$

in which the quantity B is given by

$$B = C\gamma^{2/3}\kappa^{1/3}K^{-1}(\rho_w/\rho_a)^{1/2}. \qquad (6)$$

It is convenient to introduce the friction coefficient C_f ($= u_*/\bar{u}$), and so to obtain

$$(\Delta Q/Q) = \rho_a c_p(D_H/C_f)(1 + s/\gamma_a)B + 4\epsilon\sigma T_s^3 B/u_*. \qquad (7)$$

Eq. (7) expresses the fractional increase in heat transfer rate that will accompany the complete destruction of the subsurface thermal laminar layer. The first term on the right of Eq. (7) is largely independent of wind speed, while the second is essentially inversely proportional to it. It is simple to show that, in most circumstances, the terms will be of roughly similar magnitude. The quantity D_H/C_f is the ratio of two transfer coefficients, which are stability dependent to nearly the same extent; thus the earlier neglect of the variation of transfer coefficients with surface temperature is not as critical as might have seemed. Temperature dependencies of the critical quantities in Eq. (7) are as listed in Table 1; these evaluations have been derived from published tabulations such as the *Smithsonian Meteorological Tables*. Fig. 1 shows the net effect of complete elimination of the subsurface thermal skin on the rate of total heat transfer between the water and the air. It should be emphasized that the values plotted in the diagram are subject to considerable error, on the one hand, because of the complete neglect of fog and on the other because of the need to assume particular values for the coefficients D_H and C_f. On the basis of experience at sea and at the Dresden cooling pond (see Hicks *et al.*, 1977), these have been taken to be 0.0015 and 0.035, respectively, for purposes of the present illustrations.

3. The aspirated heat loss

The method by which destruction of the subsurface thermal laminar layer is achieved is unimportant, but if the method employed involves aeration

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[3] See Appendix for list of symbols.

TABLE 1. Evaluations of the major properties controlling the potential improvement in thermal characteristics associated with sparging of a hot water body. The quantities B and A are defined by Eqs. (6) and (15), respectively. At low temperatures, values of A are difficult to estimate because of the importance of infrared sky radiation. The estimates in parentheses are therefore presented with little confidence.

Temperature (°C)	Thermal conductivity K (W m^{-1} °C^{-1})	Viscosity γ (cm^2 s^{-1})	B $\Delta T_s u_*/Q$ [°C cm s^{-1} (W m^{-2})$^{-1}$]	A (estimated) $\epsilon\sigma T_s^4/(H + L_w E)$
5.	0.565	0.01512	0.0225	(0.5)
10.	0.578	0.01304	0.0200	(0.5)
15.	0.588	0.01137	0.0181	(0.5)
20.	0.599	0.01002	0.0164	(0.5)
25.	0.607	0.00890	0.0150	(0.5)
30.	0.615	0.00797	0.0138	(0.5)
35.	0.624	0.00719	0.0128	(0.5)
40.	0.632	0.00653	0.0119	(0.5)
45.	0.638	0.00596	0.0111	0.4
50.	0.645	0.00547	0.0104	0.3
55.	0.649	0.00504	0.0098	0.3
60.	0.653	0.00466	0.0093	0.2
65.	0.659	0.00433	0.0088	0.1
70.	0.666	0.00404	0.0083	0.1
75.	0.668	0.00378	0.0080	0.1
80.	0.670	0.00355	0.0076	0.1
85.	0.672	0.00334	0.0073	0.1
90.	0.674	0.00315	0.0070	0.1
95.	0.676	0.00298	0.0067	0.1
100.	0.678	0.00282	0.0065	0.1

or sparging at sufficiently high rates, then a further heat loss mechanism should be considered. Let us assume that the air employed in the sparging process has much the same temperature and humidity as air flowing over the water surface, and that the rising bubbles are sufficiently small (or rise sufficiently slowly) that they come into thermal equilibrium with the water and become saturated at water temperature (which is assumed to be constant). It is assumed that this process eliminates the surface thermal "skin" and that as a result the bulk water temperature is T_s. If the sparging rate corresponds to an average updraft velocity w_0, then the heat used in warming the rising air bubbles will be

$$Q_{a1} = \rho_a c_p w_0 (T_s - T_a) \tag{8}$$

and that associated with saturating the air will be

$$Q_{a2} = \rho_a L_w w_0 (q_s - q_a). \tag{9}$$

The sum of Q_{a1} and Q_{a2} determines the total increment in heat loss resulting from the aspiration:

$$\begin{aligned} Q_a &= Q_{a1} + Q_{a2} \\ &= \rho_a c_p w_0 [T_s - T_a + (q_s - q_a)/\gamma_a]. \qquad (10) \end{aligned}$$

A simplification can be achieved by introducing the air wet-bulb temperature T_w, from which q_a can be evaluated as

$$q_a = q_s(T_w) - \gamma_a (T_a - T_w), \tag{11}$$

where $q_s(T_w)$ is the saturated specific humidity at temperature T_w. Combining Eqs. (10) and (11) leads to

$$Q_a = \rho_a c_p w_0 \{T_s - T_w + [q_s(T_s) - q_s(T_w)]/\gamma_a\}. \tag{12}$$

For convenience, the quantity $q_s(T_s) - q_s(T_w)$ can be approximated by $s(T_s - T_w)$, where s is the slope of

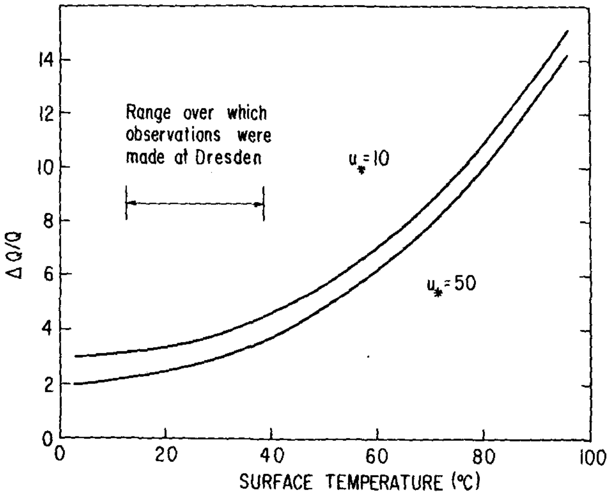

FIG. 1. The percentage increase in the total heat transfer from a water surface to the air accompanying artificial disruption of the subsurface pseudo-laminar thermal boundary layer. The curves are based on the surface layer studies of Wesely (1979). The upper curve corresponds to relatively calm conditions (2–3 m s^{-1} wind speed), while the lower corresponds to a wind of ~10 m s^{-1}; the corresponding friction velocities (u_*) are 10 and 50 cm s^{-1}.

the saturated specific humidity curve evaluated at some intermediate temperature. Then Eq. (12) simplifies immediately to

$$Q_a \approx \rho_a c_p w_0 (1 + s/\gamma_a)(T_s - T_w). \qquad (13)$$

Comparison with the total heat transfer rate from the surface, Q as expressed by Eqs. (2) and (3) leads to an estimated incremental heat loss due to aspiration

$$(Q_a/Q) = (w_0/D_H\bar{u})(1 + A)^{-1}. \qquad (14)$$

Here, A is the ratio of the net radiative to convective total heat loss from the water surface and is clearly a strong function of temperature and other meteorological variables. Some estimates of the quantity A are listed in Table 1. The tabulated values are based on the simple model of heat loss from a strongly heated water surface, presented earlier by Hicks (1975); insolation and sky radiation are assumed to be small (i.e., $R\downarrow$ is neglected) and thus the evaluations might be overestimates and should be applied with appropriate caution. These overestimates become especially large at low temperatures where in practice values exceeding 0.5 seem unlikely to occur.

By introducing the friction coefficient, it can be shown that

$$(Q_a/Q) \approx 25(w_0/u_*)/(1 + A), \qquad (15)$$

where values of 0.0015 and 0.035 have been used for D_H and C_f, as before.

4. Some experimental verification

Two sets of experiments have been conducted in order to investigate the effects of sparging on the heat transfer characteristics of a water surface. The first study used a pair of "cooling pond simulators," each about 10 m^2 in area and exposed in the natural surroundings of a grassy field at Argonne National Laboratory. The ponds were constructed of expanded polystyrene foam blocks ~10 cm thick so that conductive losses of heat through the walls were small. Each pond was lined with a black polyethylene sheet in an attempt to simulate the radiative properties of deep water and was filled with clean, filtered water to a depth of ~10 cm. Electrical heater cables laid along the bottom of each pond generated heat at known adjustable rates up to ~5 kW. In one cooling pond simulator, a series of sunken perforated pipes allowed air to be ejected in fine streams of small bubbles along rows spaced ~20 cm apart. Comparisons were then made between the temperatures of the equally heated aerated and unaerated surfaces.

The second experiment was performed with a much smaller (10 cm × 30 cm, 7 cm deep) insulated water reservoir contained in the test section of a wind tunnel. The floor of the tunnel was faired into the top of the reservoir to minimize edge effects. Vigorous aeration was employed, sufficient to ensure that the surface was uniformly covered by bubbles. Instead of employing artificial heating, the time rate of change of the temperature of initially warm water was monitored for a range of aeration rates and wind tunnel velocities. Provided that the temperature range is sufficiently small, the time constant of the exponential decay of water temperature toward ambient wet bulb temperature can be written as

$$\tau_0 = (\rho_w/\rho_a)(c_w/c_p)(h/D_H)/[u(1 + s/\gamma_a)]. \qquad (16)$$

This equation is derived directly from Eq. (3), as a result of relating the heat loss Q to the rate of change of water temperature with time. Obviously, the effects of heat loss through the walls of the container and of radiation have been neglected. Comparison between time constants appropriate for aerated and unaerated surfaces at particular wind tunnel velocities yields estimates of the corresponding apparent enhancement of the transfer coefficient D_H. The fractional increase in transfer coefficient can be alternatively interpreted as either a measure of the change in surface temperature elevation $T_s - T_w$ for a fixed heat loading or as an improvement in the rate of heat transfer at a given surface temperature.

In Fig. 2, values of the quantity $\Delta Q/Q$ deduced from determinations of $\Delta\tau_0/\tau_0$ are plotted as a function of the normalized aeration rate w_0/u_*. The friction velocity was not measured directly in either of the experiments reported here; for the present purposes values of u_* have been estimated as 3% of the wind speed at 1.5 m height in the case of the simulator studies, and as 5% of the wind speed in the wind tunnel study. The wind tunnel data, corresponding to rather high aeration rates, do indeed show the dependence on w_0/u_* that is indicated by Eq. (15). The line drawn by eye through the wind tunnel observations indicates that the fractional increase in heat transfer is roughly proportional to w_0/u_*. The constant of proportionality indicates a value of $A \approx 1$. Such a high value of A is not disturbing, since radiative effects in the case of the wind tunnel studies are difficult to quantify.

At the lower sparging rates of the simulator studies, the observations indicate little dependence on w_0/u_*, which would result if the only effect were the elimination of the thermal "skin" as predicted by Eq. (7), but the magnitude of the heat transfer enhancement appears to be substantially greater than expected (cf, Fig. 1; simulator water temperature were usually much less than 20°C). The reason for the discrepancy is not clear; it appears that the subsurface thermal boundary layer of the unaerated cooling pond simulators was substantially more prominent than in the case at the Dresden lake where the thermal "skin" observations were made,

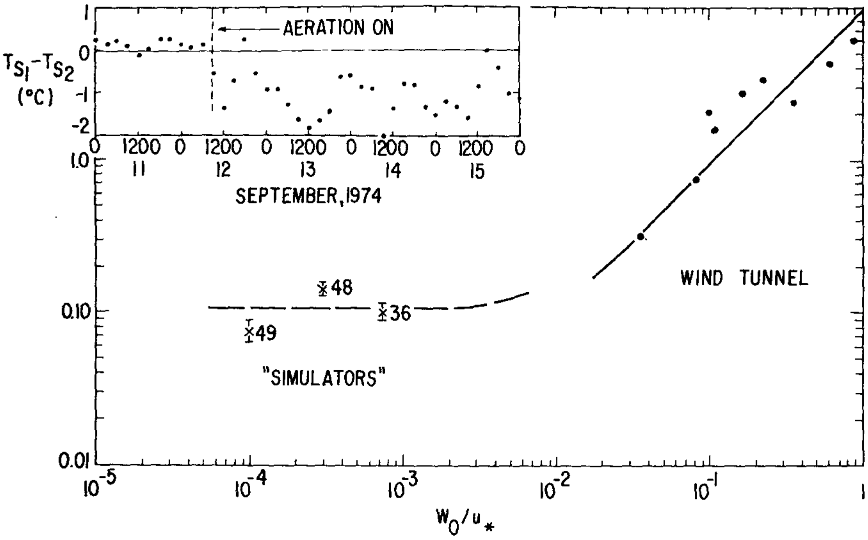

FIG. 2. The fractional increase in the total heat transfer accompanying an aeration ratio w_0, as determined experimentally from wind tunnel and simulated cooling pond investigations at Argonne. The inset illustrates the cooling that typically accompanies the commencement (in simulator 1) of aeration.

even though heating rates and friction velocities were similar. It is tempting to relate the difference to some corresponding difference in wave structure, perhaps associated with the depth of the water, but there is no convincing evidence to support this hypothesis. It is also possible that the surface of the simulators was somewhat sheltered, so that the data obtained with them reflect the influence of edge effects.

Several convincing examples of the consequences of sparging were encountered during the cooling pond simulator studies at Argonne. The inset in Fig. 2 illustrates an example of the temperature record obtained when aeration of a heated pond was commenced. Before aeration, the two simulators (identified by subscripts 1 and 2) were dissipating the same amount of heat and contained water at the same temperature. At about 1000 LST 12 September, aeration was turned on in simulator number 1. No other changes were made. On the average, water in the aerated simulator was subsequently ~1°C cooler than that in the unaerated pond.

On a number of other occasions, clearly visible steam fog originated from the aerated surface when aeration commenced, presumably as a direct consequence of a sudden increase in temperature of the water surface as the subsurface thermal skin was destroyed, and the associated increase in mixing ratio of the air just above the surface.

5. Conclusions

The benefits of sparging as a supplementary cooling pond heat dissipation method appear to be strongly related to water temperature. At 80°C, for example, Fig. 1 shows that destruction of the subsurface thermal boundary layer would allow ~10% more heat transfer without any significant increase in water temperature. With the value of A being quite small at this temperature, Eq. (15) indicates that a further 10% improvement would be attained by the use of a sparging rate of $\sim 0.004u_*$. These two heat loss increments are additive and hence in typical winds we would expect to achieve a net 20% improvement in performance from an aeration rate corresponding to a spatially averaged updraft velocity $w_0 \approx 0.1$ cm s^{-1}. At 20°C, however, destruction of the subsurface thermal skin would result in only about a 3% improvement in thermal efficiency and the 0.1 cm s^{-1} sparging rate might add another 5% or so. This indicates that the practical benefits of sparging as a heat transfer improvement mechanism might well be confined to the case of very hot water bodies, such as might be associated with emergency cooling systems.

Finally, it should be emphasized that all of the arguments presented above apply to the case of no steam fog formation. The simulator experiments conducted at Argonne yielded several occasions during which heavy fog over the surface limited the loss of longwave radiative heat; in such circumstances, the use of any method to promote heat exchange with the air by modifying the subsurface thermal boundary layer might well prove to be self-defeating.

Acknowledgments. The studies of sparging were conducted to test the intuitive and conceptual models that resulted from early discussions with

Dr. P. Frenzen. Dr. C. M. Sheih carried out the wind tunnel studies and Mr. G. A. Zerbe conducted the cooling pond simulator investigations. The field program conducted at the Dresden power plant was made possible with the cooperation of the Commonwealth Edison Company.

APPENDIX

List of Symbols

A	ratio of infrared radiative to turbulent heat loss
B	subsurface thermal film thickness parameter
c_p	specific heat of air at constant pressure
c_w	specific heat of water
C	constant
D_H	bulk transfer coefficient for sensible heat
E	evaporation rate
H	sensible heat flux
h	depth of water
K	thermal conductivity of water
L_w	latent heat of vaporation of water
Q	total heat transfer
Q_a	aspirated heat
q_a	air specific humidity
q_s	specific humidity at the surface
$R\uparrow$	outgoing radiation
$R\downarrow$	incoming radiation
s	slope of the saturated specific humidity curve
T_a	air temperature
T_s	surface temperature
ΔT_s	temperature drop across the subsurface thermal laminar layer
T_w	wet-bulb temperature
$\bar{u}$	wind speed
u_*	friction velocity
w_0	average aeration rate
γ	kinematic viscosity of water
γ_a	psychrometric constant $[=c_p/L_w]$
ϵ	emissivity (~0.99 for water)
κ	thermal diffusivity of water
ρ_a	density of air
ρ_w	density of water
σ	Stefan-Boltzmann constant
τ	momentum flux
τ_0	time constant

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