Experimental Investigation on Bubble Cooling for Large Amount of Heated Waste Water

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ABSTRACT

The objective of this paper is the study of the cooling of a large amount of heated water. Water flowing in a stream channel was cooled by air bubbles. The experimentally determined overall coefficients of enthalpy transfer between heated water and air bubbles were correlated with the flow rates of air and heated water, the results demonstrated that this cooling system could be used for cooling of large amounts of waste water with somewhat better performance than that obtained with a cooling tower.

Introduction

Large amounts of waste heat being discharged into natural water boddies (rivers, lakes, ponds or the ocean) from chemical plants or electric power plants creates a serious environmental problem; hence, the cooling of the waste water becomes important. Cooling towers and evaporative coolers are widely used for water cooling. They are not suitable, however, for a large amount of heated waste water because of the requirement of much too high capital costs.[2] Though a spray cooling method can be used, it is not suitable because its cooling performance depends on velocity and direction of wind at the water surface. Much mist is carried by the wind which causes secondary pollution, when applied to sea water.[9]

The present paper deals with a cooling method using air bubbles, and presents the fundamental aspect of the evaporative cooling in the channel through which waste water is discharged from some plant. The overall coefficients of enthalpy transfer between heated water and air bubbles were determind experimentally, and correlated with flow rates of both air and water

Experimental apparatus and procedure

The schematic diagram of the experimental apparatus is shown in Fig. 1. Heated water was circulated through a test section (air-water contact section) (7), 1 m long, 0.3 m wide and 0.3m deep, by pump (2). The temperature of the water was controlled in a reservoir (1). The flow rate was adjusted by a gate valve (3) and measured by an orifice meter (4). A perforated plate (6) was installed at the inlet of the flume to obtain more uniform water flow. The temperature of the flowing water were measured by two thermocouples at the inlet, and two at the outlet of the test section. The temperature distribution of the water in the tast section was obtained by a traversing thermocouple.

Air was blown by an air blower (9) into the water, in the test section, from ten perforated tubes of diameter 0.02m (15) through air valves (10), (11) and orifice meter (12). The perforated tubes, each with a line of holes of 0.003m diameter, drilled at intervals of 0.02m were located on the bottom of the test sectoin, at intervals of 0.1 m at right angles to the center line of the channel. The individual flow rate from each tube was controlled by the valves (16). The temperature and the humidity of the air were measured by Assmann-hygrometer at the inlet and the outlet of test section.

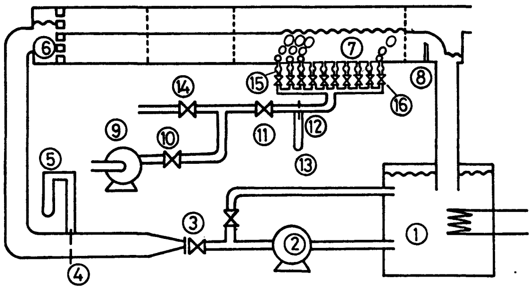

Fig. 1 Schematic diagram of experimental apparatus 1. constant temperature reservoir 2. pump for water 3, 11, 14. valves 4, 12. orifice meters 5, 13. manometers 6. perforated plate 7. test section 8. weir 9. blower 15. perforated pipe 16. adjusting valves for air flow rate

Calculation of overall enthalpy transfer coefficient

It is difficult to know the exact flow pattern of water due to the complicated behaviour of bubbles and water in the test section. For the analysis of the complicated enthalpy transport phenomena between bubbles and water, gas and liquid phases are assumed to be continuously in contact as they pass through the test section. The equations of heat balance and heat transfer rate are expressed as follows:

$$-L\,C_L \frac{\partial T}{\partial x} = G \frac{\partial i}{\partial y} \tag{1}$$

$$G \frac{\partial i}{\partial y} = K_{OG}a\,(i_L - i) \tag{2}$$

where i_L is the saturated enthalpy at the water temperature T. If the epuilibrium line is assumed to be straight, so that $di_L/dT = m$ is constant, Eq. (1) reduce to

$$-\frac{L\,C_L}{m} \frac{\partial i_L}{\partial x} = G \frac{\partial i}{\partial y} \tag{3}$$

Eliminating the variable i from eqs. (2) and (3), one can obtain

$$\frac{\partial^2 i_L}{\partial x \partial y} + \frac{K_{OG}a}{G} \frac{\partial i_L}{\partial y} + \frac{K_{OG}a\ m}{L\,C_L} \frac{\partial i_L}{\partial y} = 0 \tag{4}$$

$$x = 0,\ 0 \leqq y \leqq Y : i_L = i_{L1}\ (\text{or } T = T_1) \tag{5}$$

$$y = 0,\ 0 \leqq x \leqq x : i = i_1 \tag{6}$$

The distribution of t_L can be found whereafter i is obtained from eq. (3) by differentiation of t_L with respect to x. The solution appears as a definite integral of the product of an exponential function and Bessel function. A convinient way, however, to express the final result is that customarily used for crossflow heat exchanger calculations. That is, the variables i_L and i correspond to the temperatures of two fluids flowing perpendicularly to each other in a cross flow heat exchanger. Bowman et al[1] gives a graph showing the correction factor F to be applied to the logarithmic mean driving force to account for the cross flow conditions, as a function of

$$P = \frac{\bar{i}_2 - i_1}{i_{L1} - i_1} = \frac{1}{R}\,\frac{i_{L1} - \bar{i}_{L2}}{i_{L1} - i_1} \quad \text{and} \quad R = \frac{i_{L1} - \bar{i}_{L2}}{\bar{i}_2 - i_1}$$

where the subscript 2 and the overbar denote the exit condition and the averaged value over the cross section, respectively. Using the overall heat balance equation,

$$Q = \frac{L\,C_L Y}{m}\,(i_{L1} - \bar{i}_{L2}) = GX\,(\bar{i}_2 - i_1) \tag{7}$$

one can rewrite the parameter R as

$$R = \frac{Gm}{LC_L}\,\frac{X}{Y} \tag{8}$$

The heat transfer rate can be written as

$$Q = K_{OG}a\ XY\ F(P, R)\,\frac{(i_{L1} - \bar{i}_2) - (i_{L2} - \bar{i}_1)}{\ell n\,\dfrac{i_{L1} - \bar{i}_2}{\bar{i}_{L2} - i_1}} \tag{9}$$

Thus one can calculate $K_{OG}a$ by substituting the measured values into the followig equation:

$$K_{OG}a = \frac{1}{\dfrac{mX}{LC_L} - \dfrac{Y}{X}}\ \frac{1}{F(P, R)}\ \ell n\,\frac{1 - P}{1 - PR} \tag{10}$$

Experimental results

Typical examples of the temperature distributions of water in the x and y-direction in the bubble-water contact section (test section) are shown in Fig. 2 and 3 respectively. The variations of the water temperature in the y-direction are small as shown in Fig. 2. This means that the present enthalpy transfer process is similar to the case of single passes, one fluid mixed and other one unmixed in a cross flow heat exchanger. Accordingly, the correction factor F in Eq. (10) are taken from the graph for that case. As seen in Fig. 3, the water temperature decreases almost linearly with the distance from the inlet, supporting the assumption

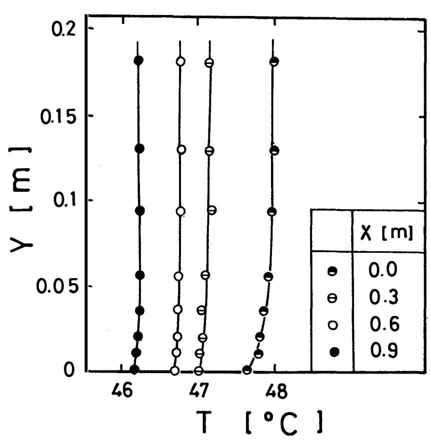

Fig. 2 Temperature distribution of water in y-direction

of uniform contact of water and gas bubbles.

The correlation of $K_{OG}a$ were made by changing water flow rate L and air bubble flow rate G at the constant depth of water flow. Figure 4 presents the experimental results which indicate that $K_{OG}a$ increases with increasing flow rates of both water and gas. From this figure, $K_{OG}a$ is found to vary proportionally to $G^{0.54}$.

In order to examine the effect of the depth of flowing water, $K_{OG}a$ was determined in the range Y between 0.084 and 0.156m, and the result is shown in Fig.5. The fact that $K_{OG}a$ is independent on Y, as shown in Fig.6, indicates that the end effect for non-uniform contact of the air bubbles and the water in the region close to the air nozzles is negligibly small in this experimental apparatus.

Ten perforated tubes are placed at intervals of 0.1m on the bottom of the stream channel, that is, x is 0.9m. In order to examine the effect of x on $K_{OG}a$, X was varied by removing the tubes from both ends in the test section. The experiments were carried out for four values of x, 0.9, 0.7, 0.5, 0.3m retaining a constant flow rate L, and a constant water depth Y. The result is shown in Fig.6. It is seen in Fig.6 that the values of $K_{OG}a$ do not depend on the number of the perforated tubes; They are independ of the length X of the cotact section. This result

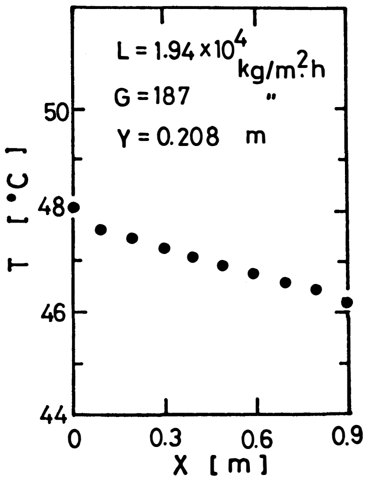

Fig. 3 Temperature distribution of water in x-direction

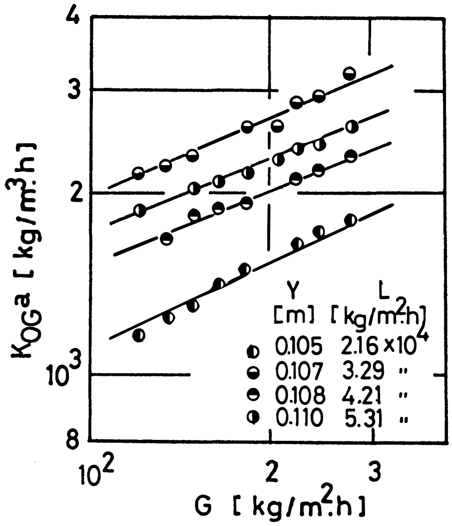

Fig. 4 Variation of $K_{OG}a$ with G and L at constant Y

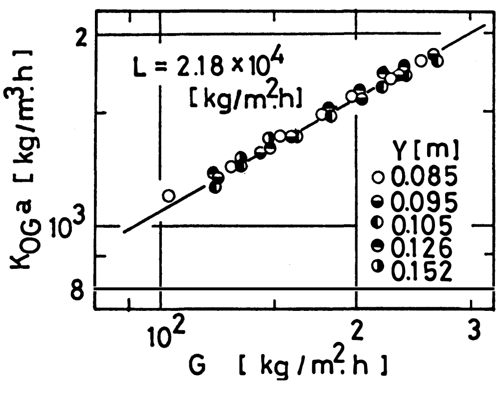

Fig. 5 Effect of depth of water flow on $K_{OG}a$

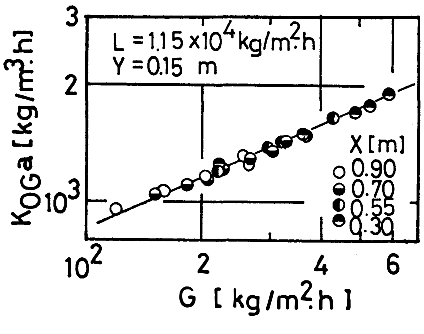

Fig. 6 Effect of length of contact section on $K_{OG}a$

is an indirect evidence of uniform distribution of air bubbles.

Since $K_{OG}a$ is found to be a function of flow rates of water and bubbles as shown in Figs. 4 to 6, values of $K_{OG}a/$ ${}^{0.54}$ are plotted against L in Fig. 7. Though the data are somewhat scatterd, $K_{OG}a$ is approximately proportional to $L^{0.5}$; the following equation is obtained:

$$K_{OG}a = 0.67\,(L)^{0.5}\,(G)^{0.54} \qquad (11)$$

Equation (11) is valid under the following experimental conditions,

$1.3 \times 10^4 < L < 9.5 \times 10^4$	kg/m^2.h
$1.1 \times 10^2 < G < 6.0 \times 10^2$	kg/m^2.h
$0.084 < Y < 0.156$	m
$0.3 < X < 0.9$	m
$40 < T < 50$	℃

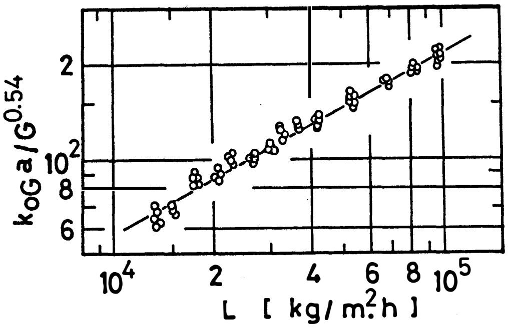

Fig. 7 Correlation of $K_{OG}a/G^{0.54}$ with L

Discussion

The overall enthalpy transfer resistance $1/K_{OG}$ is expressed as the sum of resistances in water and in air respectively:

$$\frac{1}{K_{OG}} = \frac{m}{h_L} + \frac{C_H}{h_G} \qquad (12)$$

where m is the slope of the equilibrium curve of saturated enthalpy versus temperature, the heat capacity of air in the bubble, h_L and h_G the water and air side heat transfer coefficients respectively. In deriving eq. (12), the Lewis relation is used.

Very few data have been published for the studied cooling device; this cross flow-condition might be similar to that of the flows on trays in bubble-cap or sieve-plate columns; for these, some mass transfer have been presented in terms of plate efficiency.[3,(8)] Unfortunately, it is difficult to compare our heat transfer data with these mass transfer data because the empirical results for mass transfer do not account for several significant variables, including fluid velocities, tray dimensions, and some physical properties.

Gerster et al.[4)] obtained the empirical equation for gas- and liquid-phase mass transfer resistances for several tray designs. Assuming the analogy between heat and mass transfer, one can calculate the values of m/h_La and C_H/h_Ga from their empirical relations. The calculated result for $L=4\times10^4$ and $G=200$ kg/m^2.h shows that the gas phase resistance predominates and the calculated value of the overall coefficient $K_{OG}a$ is about 25% higher than that obtained in the present experiment. However, the dependency of the calculated $K_{OG}a$ on L and G does not agree with the present result. Therefore, it is neccesary to execute more detailed exqeriments in order to obtain better

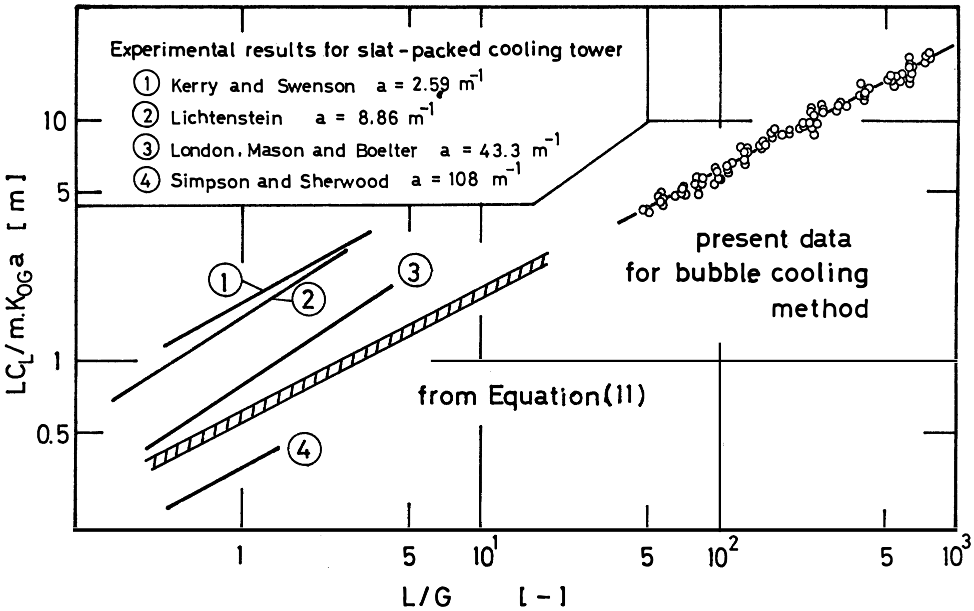

Fig. 8 Comparison with experimental values of height of transfer unit for slat-packed cooling towers

understanding the transfer mechanism between bubble and water flows in such a cooling system. Though the present experimental conditions differ from those in cooling towers, the results are compared in Fig. 8 with the experimental values for slat-packed cooling towers from various sources.[5,6,7,10] All of the present data exist in the region of very large values of L/G. If we assume that the present results can be extended to the region of small values of L/G, the value of $K_{OG}a$ for the bubble cooling is somewhat larger than those of kerry et al., Lichtenstein and London et al., and smaller than that of Simpson et al. That is, the present cooling method can be used for the cooling of a large quantities of water with somewhat better performance than that obtained with cooling towers.

Conclusion

To cool a large amount of discharged waste water, bubble cooling method was studies.

The overall enthalpy transfer coefficient $K_{OG}a$ obtained by the experiments is independent of depth of water and length of the contact section. The dependence on L and G can be represented by the empirical equation:

$$K_{OG}a = 0.67\ (L)^{0.54}(G)^{0.5}$$

in the ranges $1.3\text{x}10^4 < L < 9.5\text{x}10^4$ kg/m^2.h, $110 < G < 600$kg/m^2.h, $0.084 < Y < 0.156$ m, $0.3 < X < 0.9$m, $45 < T < 50$℃

It was observed that this cooling system could be effectively used for cooling of a large amount of waste water with somewhat better performance than that involving a cooling tower.

Acknowledgment

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Nomenclature

a = effective interfacial area 〔m^2/m^3〕
C_L = heat capacity of water 〔J/kg.K〕
F = correction factor 〔−〕
G = mass velocity of bubbles based on channel surface area 〔$kg/m^2.h$〕
i = enthalpy of air 〔J/kg-dry air〕
i_L = saturated enthalpy of air at water 〔J/kg-dry air〕
= temperature T 〔J/kg-dry air〕
$K_{OG}a$ = overall coefficient of enthalpy transfer 〔$kg/m^3.h$〕
L = mass velocity of heated waste water based on cross sectional area of the channel 〔$kg/m^2.h$〕
m = slope of saturated enthalpy-temperature curve 〔J/kg.K〕
P = function of correction factor 〔−〕
R = function of correction factor 〔−〕
T = temperature of the water 〔℃〕
t = temperature of the air 〔℃〕
X = length of the contact section 〔m〕
v = longitudinal coordinate 〔m〕
Y = depth of the water flow 〔m〕
y = vertical coordinate 〔m〕
Z = width of channel 〔m〕
z = transverse coordinate 〔m〕

Subscripts

1 : inlet
2 : outlet

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