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Heat and mass transfer from a horizontal tube of an evaporative heat dissipator

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Abstract—In this investigation, the mass transfer coefficient with simultaneous water and air flow has been determined experimentally. The coefficient has also been calculated theoretically from the convective heat transfer coefficient of air by applying the Lewis relation for an air–water mixture. The ratio of experimental to theoretical mass transfer coefficients has been found to lie between 0.80 to 9.35. A new term 'evaporative effectiveness' is defined as the ratio of energy dissipated in evaporative cooling to that in simple water cooling. Its variation is studied and found to range from 0.85 to 1.78. Correlations for design purposes are recommended in this paper.

INTRODUCTION

EVAPORATIVE tubular heat exchangers are employed widely for cooling of hot process fluid. Such a situation is found in chemical industries, refrigerant condensers, etc. In an evaporative tubular heat exchanger, the water is sprayed over horizontal tubes inside which hot fluid, which has to be cooled, passes. Simultaneous heat and mass transfer between the water and circulating air occurs inside the equipment and the energy transfer depends upon these mechanisms. Most of the earlier work [1–7] on evaporative heat exchangers considered a nest of tubes in the form of a coil as a model for their experimental investigation. In order to understand the mechanism of heat and mass transfer in such units, it is considered desirable to investigate first the mechanism of the process. This study has been done on the basis of the behaviour of a single tube whose performance was then compared with that of a row of the tubes or banks of tubes [8].

The mass transfer coefficient with simultaneous air and water flow is normally calculated on the basis of enthalpy difference between saturated air at the tube surface temperature and air flowing in the test unit as given by:

$$Q = KA_0(i_{s,t_c} - i_{a_1}). \tag{1}$$

Rana and Charan [9] have studied the effect of Reynolds numbers of process fluid, cooling water and air on this coefficient for a single horizontal tube. They have given an empirical relation for the mass transfer coefficient in terms of these parameters. The study of the effect of the process fluid Reynolds number (Re_p) is useful from a design point of view. However, this parameter includes the effect of physical properties and flow details of process fluid. In order to eliminate their effect, the use of enthalpy potential, Δi, was attempted:

$$\Delta i = i_{s,t_c} - i_{a_1}. \tag{2}$$

The effect of wall temperature has been substituted for the process fluid Reynolds number. In fact, the wall temperature affects the enthalpy potential, which is widely believed to affect the simultaneous heat and mass transfer during evaporative cooling.

It is a general practice that the mass transfer coefficient with simultaneous water and air flow is also theoretically predicted from the heat transfer coefficient of the test unit with only air flow by using the Lewis relation for an air–water mixture [7] given by:

$$K = h_a / Le\, C_a. \tag{3}$$

McAdams [10] has recommended correlations for the mean heat transfer coefficient of air for a single isothermal cylinder for different ranges of Re_a. Zukauskas [11], on the basis of his experimental data, has also obtained correlations for mean heat transfer coefficient of a single, isothermal tube in terms of Re_a alone. Rana and Charan [12] studied the additional effect of Re_p passing through the tube. They have observed that the Nusselt number decreases slightly with Re_p. They too have given the empirical relation for Nusselt number in terms of Re_a alone, having ignored the effect of Re_p as negligible. Further, they used their correlation for calculating the heat transfer coefficient with only air flow to predict the theoretical mass transfer coefficient by using Lewis relation. Rana [8] recommends that the correlation developed by Rana and Charan [12] should not be used under conditions other than those prevailing in their investigation. He has clearly stated that they had empirically correlated their experimental data with a view to obtain a convenient relation to determine heat transfer coefficient for the conditions of their investigation as required in the further development of the analysis.

THE TEST FACILITY

The test facility used in this investigation is shown schematically in Fig. 1. The test unit, 1, was made from

NOMENCLATURE

A	surface area [m^2]
C	specific heat at constant pressure [J kg^{-1} K^{-1}]
D	test unit tube diameter [m]
EE	evaporative effectiveness as defined by equation (6)
G	mass flow rate [kg s^{-1}]
i_a	enthalpy of air [J kg^{-1}]
$i_{s,tc}$	enthalpy of saturated air at average wall temperature [J kg^{-1}]
i_{fg}	latent heat of vaporisation of water at inlet temperature [J kg^{-1}]
K	mass transfer coefficient during simultaneous air and water flow [kg m^{-2} s^{-1}]
L	active length of test unit [m]
Le	Lewis number
Nu	Nusselt number
Q	heat flow rate [W]
R	ratio of experimental and theoretical mass transfer coefficients as defined by equation (11)
Re	Reynolds number, $4G/\pi D_i\mu$ for flow inside tube and $4\Gamma/\mu$ for film flow over a horizontal tube
t	temperature [K]
v	velocity [m s^{-1}].

Greek symbols

ρ	density [kg m^{-3}]
μ	dynamic viscosity [N s m^{-2}]
Γ	mass flow rate of cooling water per side per unit axial length of the tube, $G_w/2L$ [kg m^{-1} s^{-1}]
Δt	temperature potential [K]
Δi	enthalpy potential defined by equation (2) [J kg^{-1}]
$\Delta i/i_{fg}$	dimensionless enthalpy potential.

Subscripts

a	air
c	test tube surface
exp	experimental
i	inside
max	maximum
o	outside
p	process fluid (hot water)
s	saturated
theo	theoretical
w	cooling water
wa	cooling water and air
1	inlet
2	outlet.

12.76-mm-O.D. copper tube and placed horizontally between the two opposite sides of the test section. The other two sides of the test section were made of Plexiglass sheet for visual observation of the test surface and flow pattern of the falling cooling water. The active length of the test unit on which cooling water fell was 683 mm. Hot water as process fluid was supplied to the test unit by a pump, 14, from a hot water reservoir, 12, which was fitted with heating elements, 13. The flow rate of process fluid was measured by a calibrated rotameter, 18. The flow rate of process fluid through the test unit was regulated by connecting a bypass line and desired adjustment of bypass valve, 16, in the line. On leaving the test unit, the process fluid returned to the hot water reservoir but the fluid could also be directed to a container, 21, placed on a balance, 20, for weighing the flow rate of the pumping fluid when desired. For measuring temperature of the process fluid, the test unit was provided with two thermocouples at the entry and exit of the active length. The difference between these temperatures is used in calculating the heat dissipation rate of process fluid. Three additional thermocouples were placed equidistant from entry and exit. Thermocouples were also provided on the test unit surface, at the same locations but in different radial positions, to obtain the average temperature of the wall of the tube.

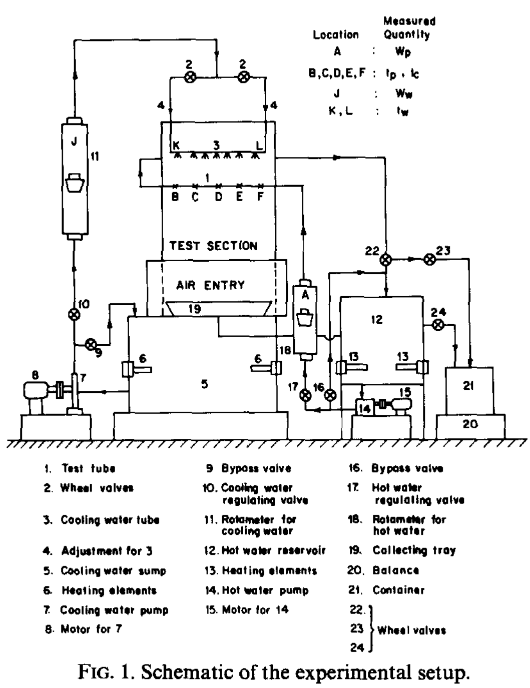

FIG. 1. Schematic of the experimental setup.

Cooling water was fed to the cooling water tube, 3, by a pump, 7, from the sump, 5, which is fitted with heating elements, 6. The cooling water tube was made of 19.05-mm-G.I. pipe whose total length was 750 mm. Holes of 1.6 mm diam., 3.18 mm apart, were drilled along the whole length of the tube. Grooves 1.5 mm deep were provided over the periphery of the tube. Each hole was drilled at the centre of the groove, so that the water flowed down without disturbing the flow emanating from the neighbouring holes. The holes were kept on the top of tube. This arrangement ensured uniform distribution of water from the tube on the test tube. Cooling water flow rate was measured by a calibrated rotameter, 11. This water subsequently fell into a tray, 19, placed in the lowest part of the test section from where it flowed into a container, 21, placed on a balance, 20. Its temperature was measured by two thermocouples located in the reducer elbows. Wheel valves, 2, regulated an equal supply of cooling water to its two paths. The air from the air washer through the air duct entered the bottom of the test section. After passing over the test unit, it left from the top of the test section where its velocity was measured. The dry- and wet-bulb temperatures of the air were measured in the air duct just above the test unit with the help of a set of dry- and wet-bulb thermometers.

TEST PROCEDURE

The mass transfer coefficient was determined with air and water flowing simultaneously. In the first set of observations, the air velocity was fixed at 0.825 m s^{-1}. The cooling water flow rate was fixed at 0.333×10^{-2} kg s^{-1} and the hot water flow rate was varied from 1.667×10^{-2} to 16.667×10^{-2} kg s^{-1} in 10 equal steps at intervals of 1.667×10^{-2} kg s^{-1}. The heat dissipated by the test unit was calculated from the flow rate and temperature difference of hot water at the entry and exit of the test tube. The outside wall temperature was recorded by thermocouples; as was the inlet temperature of cooling water. In the second set of observations, the flow rate of cooling water was varied from 0.333×10^{-2} to 1×10^{-2} kg s^{-1} in steps of 0.333×10^{-2} kg s^{-1} and from 1×10^{-2} to 6×10^{-2} kg s^{-1} in steps of 1×10^{-2} kg s^{-1}. The tests of the set 1 were repeated for each flow rate of cooling water in the set 2. Sets 1 and 2 were repeated for air velocities 1.240, 1.667, 2.083 and 2.660 m s^{-1}.

The energy dissipated by a simple, water-cooled unit was also measured. To obtain this, the procedure described above, adopted in the case of mass transfer coefficient with air and water flowing simultaneously, was repeated without air flow.

THEORETICAL

The heat dissipation rate from the tube with only water flow is given by:

$$Q_w = G_p C_p (t_{p_1} - t_{p_2})_w. \tag{4}$$

The heat dissipation rate from the tube with air and water flowing simultaneously is given by:

$$Q_{wa} = G_p C_p (t_{p_1} - t_{p_2})_{wa}. \tag{5}$$

Rana and Charan [9] defined the ratio of energies dissipated in evaporative cooling to that in simple water cooling as 'evaporative effectiveness' (*EE*); this is given by:

$$EE = \frac{Q_{wa}}{Q_w}. \tag{6}$$

It is assumed that from the wetted surface of the tube, mass transfer would take place due to the enthalpy potential given by the difference between the enthalpy of saturated air at tube average temperature and that of air at the inlet of the heat dissipator. The experimental mass transfer coefficient is then given by:

$$Q_{wa} = K_{exp} A_o (i_{s,t_c} - i_{a_1}). \tag{7}$$

The theoretical mass transfer coefficient, on the basis of heat and mass transfer anology, is calculated by applying the Lewis relation:

$$Le = \frac{h_a}{K_{theo} C_a}. \tag{8}$$

For humid air, *Le* is taken as 0.92 [7] and h_a, with air only, is given by the following correlation developed by Rana and Charan [12]:

$$Nu = 0.660 (Re_a)^{0.535}. \tag{9}$$

The Reynolds number of air is calculated as follows:

$$Re_a = \frac{D_o v_{a,max} \rho_a}{\mu_a}. \tag{10}$$

Where $v_{a,max}$ is based on the minimum free area of flow, obtained by subtracting the projected area of tube from the cross-section area of the test section and ρ_a and μ_a are evaluated at the mean air film temperature. The ratio of experimental and theoretical mass transfer coefficients is given by:

$$R = \frac{K_{exp}}{K_{theo}}. \tag{11}$$

RESULTS AND DISCUSSION

1. *Mass transfer coefficient*

The values of mass transfer coefficient under simultaneous downward flow of water and upward flow of air over the tube have been calculated from equation (7) on the basis of the difference between the enthalpy of saturated air at the tube surface temperature and that of air at inlet conditions. The mass transfer coefficient has been found to be between 5.242×10^{-2} and 0.516 kg m^{-2} s^{-1} for dimensionless enthalpy potential varying from 2.71×10^{-2} to 0.1573, the Reynolds number of cooling water varying from 9.7 to 217.4, and the Reynolds number of air varying from 744 to 2403. Figure 2 is a typical plot of mass transfer coefficient vs dimensionless enthalpy potential at different Reynolds number of cooling water and for a constant Reynolds number of air, i.e. about 1122. The mass transfer coefficient is found to decrease with increase in enthalpy potential. In this figure it is seen

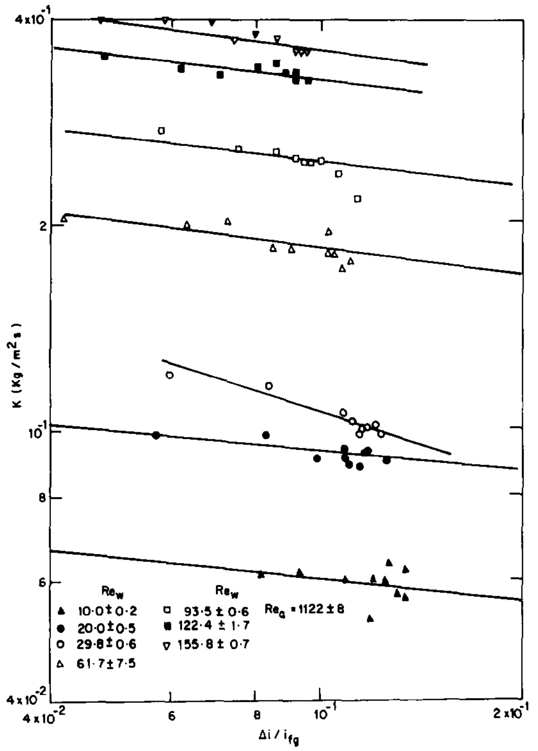

FIG. 2. Effect of $\Delta i/i_{fg}$ on K of a single tube.

that data points tend to fall on parallel, straight lines on the log–log chart.

To investigate the effect of Reynolds number of air on mass transfer coefficient, the latter was plotted vs air side Reynolds number at different values of cooling water Reynolds number and for constant dimensionless enthalpy potential of about 8.5×10^{-2}. One such plot is shown in Fig. 3. In general, it is observed that the mass transfer coefficient increases with Re_a up to about 1874 and a sharp decrease in its value is noticed as Re_a increases from 1874 to 2365. The Reynolds number of air (about 2365) corresponds to an air velocity of 2.660 m s^{-1}. It was observed that at this velocity the flowing air carries away the cooling water physically, shearing it from the tube surface which is the reason for the sharp decrease in mass transfer coefficient. This trend is exhibited for Re_w more than about 21. At Re_w lower than 21, the effect is not noticed; $Re_w = 21$ corresponds to a flow rate of cooling water of about 1.171×10^{-2} kg m^{-1} s^{-1} which is just above the minimum wetting rate. At the minimum wetting rate, the water film thickness on the tube is minimum and it adheres to the tube surface under the effect of surface tension. The kinetic energy of the air stream is not sufficient to strip water from the tube surface. As the water film thickness increases with Re_w, the possibility of its breakup under the effect of kinetic energy of on flowing air also increases. This results in an increase in the rate of cooling water carried away by air and consequently a decrease in mass transfer coefficient. The zone of sharp decrease of mass transfer coefficient is shown by dotted lines.

Figure 4 is a typical graph of the effect of Re_w on mass transfer coefficient for varying flow rates of air and a constant dimensionless enthalpy potential of about 6.5×10^{-2}. It is observed that, in general, the mass transfer coefficient increases with Re_w. The effect of Re_w is to increase the convective effect at the tube wall which increases the mass transfer coefficient. All the data points corresponding to a particular Re_a are found to be falling on a straight line. These lines for different values of Re_a appear to be parallel with the exception of that for $Re_a = 2365$. The reason for this deviation has already been discussed in the last paragraph where the effect of Re_a on mass transfer coefficient has been discussed.

The above discussion shows that it may be possible to correlate the mass transfer coefficient in terms of dimensionless enthalpy potential and Reynolds numbers of cooling water and air. However, for the purpose of development of correlation, the results for $Re_a \sim 2365$ are omitted because at this flow rate the trend of variation of mass transfer coefficient is reversed. The effect of different parameters on mass transfer coefficient was statistically analysed and the data is correlated by the following equation:

$$K_{exp} = 1.8433 \times 10^{-3}(\Delta i/i_{fg})^{-0.0652} \times (Re_w)^{0.6707}(Re_a)^{0.2506}. \quad (12)$$

The experimental values of mass transfer coefficient

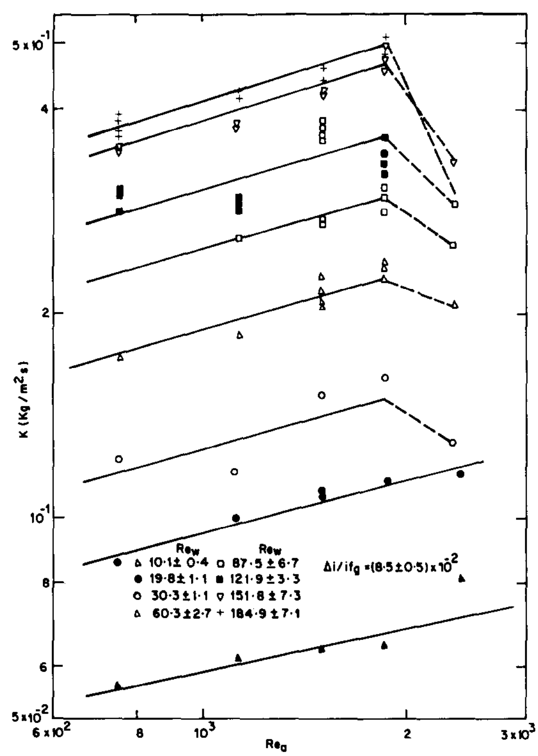

FIG. 3. Effect of Re_a on K of a single tube.

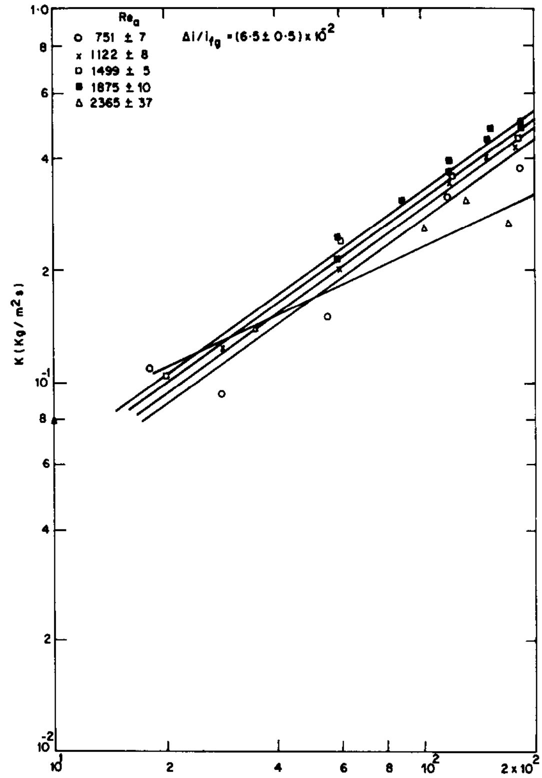

FIG. 4. Effect of Re_w on K of a single tube.

are compared with the values calculated by the above correlations as shown in Fig. 5. It is observed that for all test runs, except for five out of 320, the predicted values lie within $\pm 20\%$ of the experimental data and for 94% of the test runs, the predicted values lie within $\pm 15\%$ of the experimental values. The mean and standard deviations are 0.3371 and 8.28%, respectively. Thus the developed correlation is quite successful.

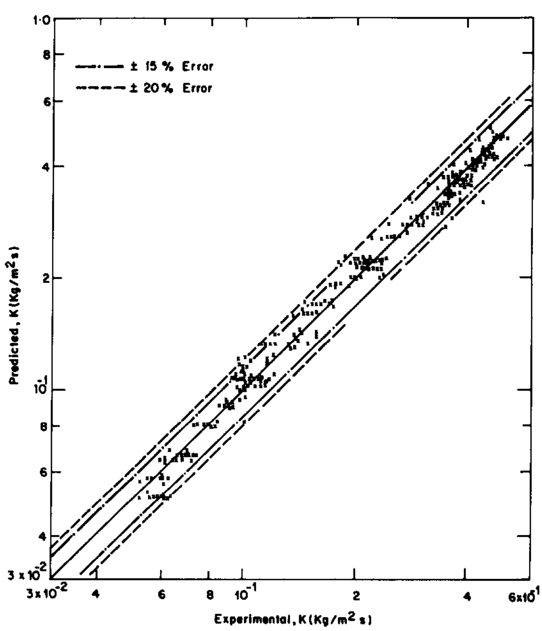

FIG. 5. Comparison between experimental and predicted K of a single tube.

2. *Ratio of experimental and theoretical mass transfer coefficients*

The theoretical mass transfer coefficient, using equations (8) and (9), is given by:

$$K_{\text{theo}} = \frac{0.66K_a(Re_a)^{0.535}}{Le\, C_a\, D_o}. \tag{13}$$

Equation (13) shows that the theoretical mass transfer coefficient is related to Re_a in terms of air properties which vary through a narrow range for the operating temperatures of this investigation. However, it is seen from equations (12) and (13) that the ratio of experimental and theoretical mass transfer coefficients, $K_{\text{exp}}/K_{\text{theo}}$, is related to Re_a, Re_w and EE by power laws. The ratio of experimental and theoretical mass transfer coefficients varies quite substantially, lying between 0.798 to 9.346 for the range of variables covered in this investigation. It was considered desirable to fit a correlation by regression techniques between R and Re_w, Re_a and $\Delta i/i_{fg}$. The developed correlation is given here:

$$R = 1.1578(\Delta i/i_{fg})^{0.0654}(Re_w)^{0.6706}(Re_a)^{-0.2691}. \tag{14}$$

The predicted values are compared with experimental values as shown in Fig. 6. It was found that for all test runs, except for five out of 320, the predicted values lie within $\pm 20\%$ of the experimental points and for 94% of the test runs, the predicted values lie within $\pm 15\%$ of experimental values. The mean and standard deviations are 0.339 and 8.31%, respectively.

3. *Evaporative effectiveness*

Figure 7 shows a typical variation of evaporative effectiveness with dimensionless enthalpy potential for constant Re_a of about 1122 for various flow rates of

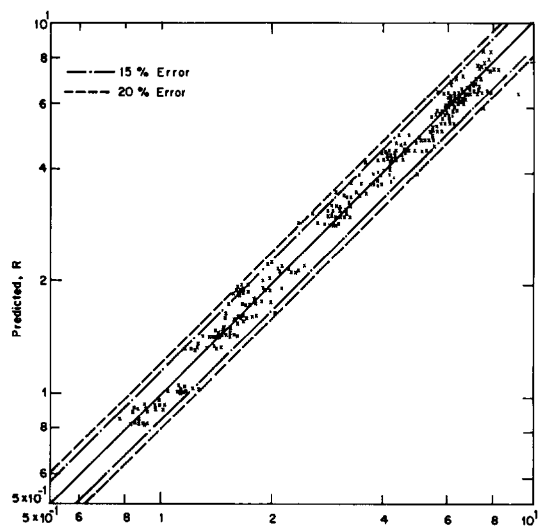

FIG. 6. Comparison between experimental and predicted R of a single tube.

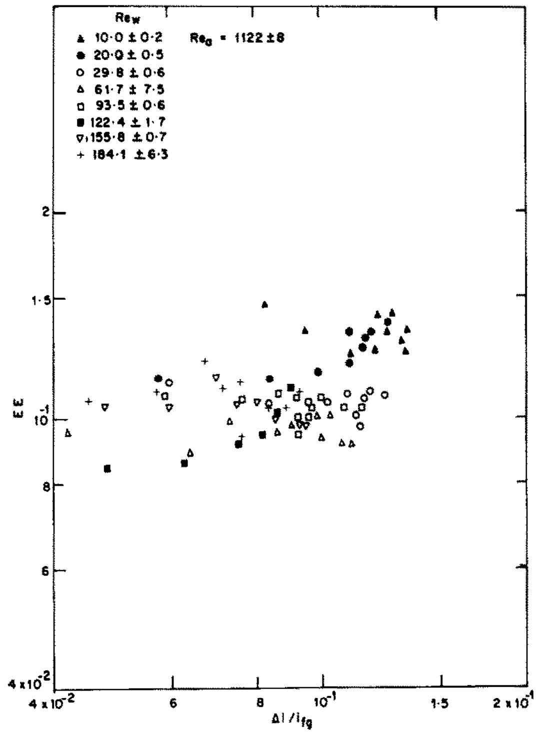

FIG. 7. Effect of $\Delta i/i_{fg}$ on EE of a single tube.

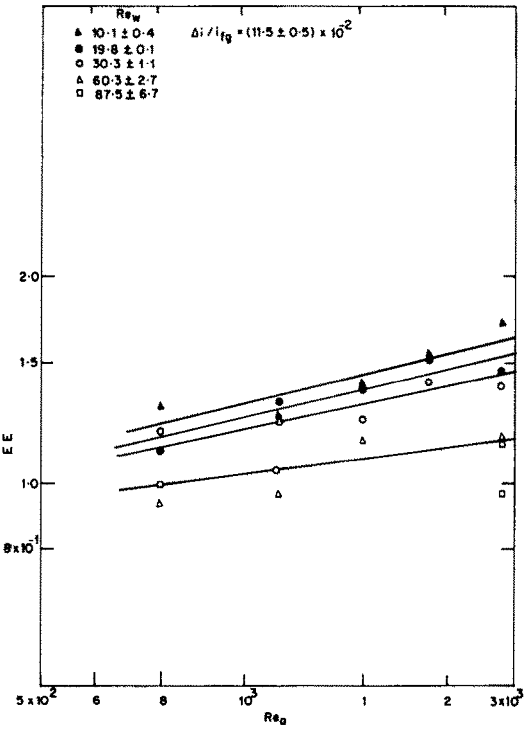

FIG. 9. Effect of Re_a on EE of a single tube.

cooling water. No specific trend is exhibited by this plot. The variation of evaporative effectiveness is random at higher enthalpy potentials. However, at low dimensionless enthalpy potentials, the evaporative effectiveness is not influenced by dimensionless enthalpy potential.

Figure 8 shows a typical plot of the variation of EE vs Re_w for different values of Re_a and at a fixed dimensionless enthalpy potential of about 0.115. It is observed that evaporative effectiveness decreases slightly as Re_w increases and the variation is linear on the log–log scale.

The evaporative effectiveness was plotted vs Reynolds number of air. Figure 9 is a typical plot of EE vs Re_a for different Re_w and for a fixed enthalpy potential of about 0.115. The evaporative effectiveness is found to increase with Re_a. It has been possible to draw straight lines to relate EE with Re_a on a log–log scale.

Thus, it is found that EE is proportional to certain exponential powers of Re_a and Re_w. The influence of these parameters was further studied by regression techniques. The variable enthalpy potential was also considered for the purpose of generality, although its

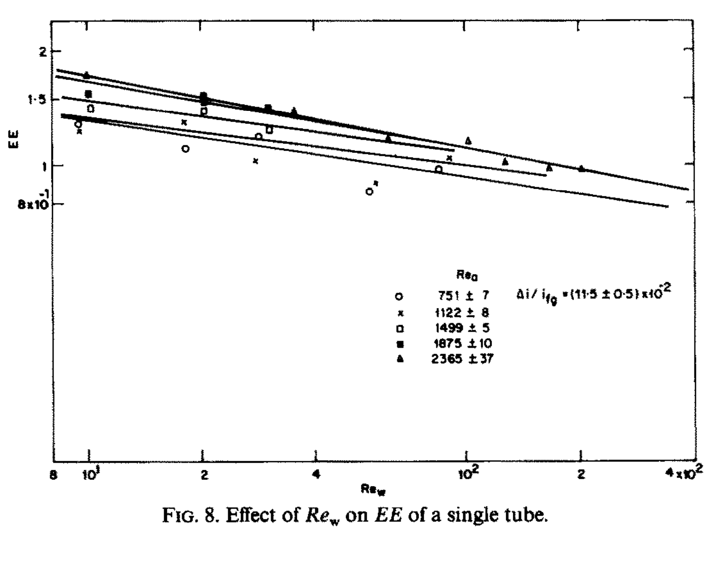

FIG. 8. Effect of Re_w on EE of a single tube.

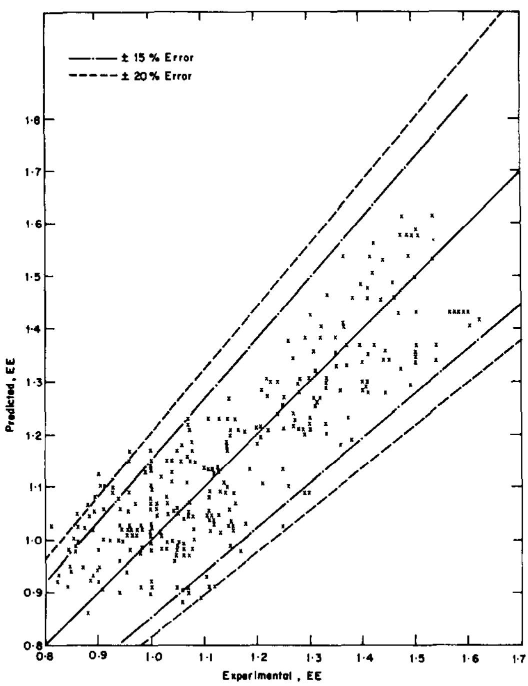

FIG. 10. Comparison between experimental and predicted *EE* of a single tube.

effect was expected to be insignificant. The following dimensionless correlation could be developed for the range of parameters investigated except for $Re_a \sim 2365$, for the reason which has been discussed earlier.

$$EE = 0.4671(\Delta i/i_{fg})^{0.0533}(Re_w)^{-0.0900}(Re_a)^{0.1916}. \quad (15)$$

The predicted values of *EE* from equation (15) are compared with experimental values in Fig. 10. It is found that for 90% of the test runs, the calculated values lie within $\pm 15\%$ of the experimental values whereas, with the exception of 14 out of 320 data points for all other test runs, the predicted values are within $\pm 20\%$ of the observed values. The mean and standard deviations are 0.508 and 10.08%, respectively.

It is noticed that in equation (15) the exponent of dimensionless enthalpy potential and Re_w are very small and for all practical purposes they can be taken as zero. The new simplified correlation in terms of Re_a only was developed by mathematical regression as:

$$EE = 0.3096(Re_a)^{0.1796}. \quad (16)$$

The predicted values of *EE* given by equation (16) when compared with experimental values were found to lie within $\pm 20\%$ for 80% of the test runs. The mean and standard deviations are 1.021 and 14.65%, respectively and are higher as compared to those given by equation (15). Comparing the agreement of equations (15) and (16) with experimental data, it is inferred that the first equation shows substantial improvement, even though the exponents of *EE* and Re_w are very small.

CONCLUSIONS

1. The mass transfer coefficient with air and water flow for a single tube in the range of $2.71 \times 10^{-2} \leqslant EE \leqslant 0.1573$, $9.7 \leqslant Re_w \leqslant 217.4$ and $744 \leqslant Re_a \leqslant 1874$ could be estimated from the following correlation

$$K = 1.8433 \times 10^{-3}(\Delta i/i_{fg})^{-0.0652} \times (Re_w)^{0.6707}(Re_a)^{0.2506}$$

2. The ratio of experimental and theoretical mass transfer coefficients of a single tube for the above mentioned range of parameters has been correlated in terms of dimensionless enthalpy potential and Reynolds numbers of cooling water and air by the following equation

$$R = 1.1578(\Delta i/i_{fg})^{-0.0654}(Re_w)^{0.6707}(Re_a)^{-0.2691}.$$

3. The evaporative effectiveness for a single tube for the range of operating parameters given above could be predicted by the following dimensionless correlations

$$EE = 0.4671(\Delta i/i_{fg})^{0.0533}(Re_w)^{-0.0900}(Re_a)^{0.1916}$$

$$EE = 0.3096(Re_a)^{0.1796}$$

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TRANSFERT DE CHALEUR ET DE MASSE AUTOUR D'UN TUBE HORIZONTAL D'UN EVAPORATEUR

Résumé—Dans cette étude, le coefficient de transfert de masse avec un écoulement simultané d'eau et d'air est déterminé expérimentalement. Le coefficient est aussi calculé théoriquement à partir du coefficient de convection thermique de l'air en appliquant la relation de Lewis pour un mélange air–eau. Le rapport du coefficient expérimental de transfert massique à celui théorique est entre 0,80 et 9,35. Un nouveau terme "efficacité évaporative" est défini comme le rapport de l'énergie dépensée dans la réfrigération évaporative à celle dans un simple refroidissement d'eau. Sa variation est étudiée et elle est située entre 0,85 et 1,78. Des formules sont recommandées dans un but pratique.

WÄRME- UND STOFFÜBERGANG AM HORIZONTALEN ROHR EINES VERDUNSTUNGSKÜHLERS

Zusammenfassung—In dieser Untersuchung wurde experimentell der Stoffübergangskoeffizient bei gleichzeitiger Strömung von Luft und Wasser bestimmt. Zusätzlich wurde der Stoffübergangskoeffizient theoretisch aus dem konvektiven Wärmeübergangskoeffizienten für Luft, durch Anwendung der Lewis'schen Beziehung auf das Luft–Wasser-Gemisch, berechnet. Das Verhältnis zwischen experimentellen und theoretischen Stoffübergangskoeffizienten lag zwischen 0,80 und 9,35. Ein neuer Term, die 'Verdunstungseffektivität', wird eingeführt als das Verhältnis aus den abgeführten Energien bei Verdunstungskühlung und bei einfacher Wasserkühlung. Die Veränderung dieser Größe wird untersucht, sie liegt zwischen 0,85 und 1,78. Korrelationen für Auslegungszwecke werden in dieser Arbeit empfohlen.

ТЕПЛО-И МАССОПЕРЕНОС ОТ ГОРИЗОНТАЛЬНОЙ ТРУБЫ ИСПАРИТЕЛЬНОГО ОХЛАДИТЕЛЯ

Аннотация—Проведено экспериментальное определение коэффициента массопереноса при одновременном течении воды и воздуха, а также его теоретический расчет по коэффициенту конвективного теплообмена для воздуха из соотношения Льюиса для смеси вода–воздух. Найдено, что отношение значений экспериментальных и теоретических коэффициентов массопереноса лежит в диапазоне от 0,80 до 9,35. Введено понятие "испарительной эффективности", определяемой как отношение диссипированной энергии при испарительном охлаждении и энергии при обычном испарении воды. Определено, что это отношение изменяется в диапазоне от 0,85 до 1,78. Даны рекомендации по использованию полученных зависимостей при расчетах.