International Journal of Thermal Technologies
ISSN 2277 - 4114

Available at http://inpressco.com/category/ijtt

Research Article

Performance Analysis of a Row of Horizontal Tubes of an Evaporative Tubular Heat Exchanger

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Accepted 10August 2013, Available online 15 Sept. 2013, **Vol.3, No.3 (Sept. 2013)**

Abstract

The results of experimental investigations on mass transfer coefficient and evaporative effectiveness of evaporative tubular heat exchanger are presented in this article. Evaporative effectiveness is estimated over a wide range of operating conditions. Based on the experiments, correlations derived using the multiple regression analysis. Developed correlations and experimental data show that as the film flow rate increases, Evaporative Effectiveness and mass transfer coefficient increases provided that the air flow rate is constant which is flowing from underneath the tubes of the evaporative tubular heat exchanger. The tubes are subjected to simultaneous flows of cooling water from the top; air flows from underneath with controlled amount of humidity and the process fluid is flowing through the tubes. Developed correlations are helpful in improvement of the design of heat transfer devices and many other engineering applications.

***Key words**: Performance Analysis, Heat Exchanger*

Introduction

Falling droplets on evaporative tubular heat exchanger, having a row of horizontal copper tubes, arranged one upon the other, thus forming a coil, have been used for more than one and a half century in refrigeration systems, power plants and chemical industries etc. for energy conversion processes as evaporators or condensers. It is a device, which employs combination of water and air to dissipate energy from a hot water flowing inside the tube. The performance of evaporative heat exchanger is generally influenced by heat and mass transfer coefficients. Higher the value of these coefficients, the greater would be the effectiveness of the evaporative tubular heat exchanger. It is necessary that the temperature of cooling water should not exceed a certain prescribed value for a particular process plant. The use of air as an external heat absorbing medium has not been adopted widely due to poor heat transfer from surface being cooled to air. Considerable increase in the rate of heat transfer between the atmospheric air and the circulating water can be achieved by bringing water into direct contact with moving water. This employs the principle of evaporative cooling of water well known since ancient times. A number of researches in the past have made both analytical as well as experimental studies to enhance the performance of evaporative tubular heat exchanger. The mechanism of falling film breakdown on an adiabatic vertical surface and subsequent rewetting of the dry patches were studied earlier (Hartley, DE *et al* 1964). This work was later improved by various investigators (Zuber, N *et al* 1966; Hodgson, JW *et al* 1968; Munakata, T *et al* 1975; Bankoff, SG *et al* 1978; Fujita, T *et al* 1978; Arefyev, KM *et al* 1979). Experimental study of the drop wise evaporation on hot surfaces was done by researches (Banacina, C *et al* 1979).

Experimental investigations to determine mass transfer coefficient and evaporative effectiveness on various test units have been reported by various researchers (Grissom, W. *et al* 1981; Perez- Blanco, H. *et al* 1984; Hallett, VA. *et al* 1996; Yasuo *et al* 1998; Pascal, *et al* 2003; Danko, G. *et al* 2006) and developed correlations. For optimal design of heat exchanger, the correlations presented by investigators (Raj Kumar *et al* 1998), have been quite useful for energy conservation in tubular evaporative heat exchangers. Analysis of a row of tubes of tubular evaporative heat exchanger has been done by researchers (Raj Kumar *et al* 2001), to get optimal evaporative effectiveness. Adiabatic film cooling effectiveness of different patterns measured using heat-mass transfer analogy method determined by researchers (Yuzhen *et al* 2006), as an advanced cooling scheme to meet increasingly stringent combustor cooling requirements. Turbulent forced convective heat and mass transfer downstream of blockages with round and elongated holes in a rectangular channel was studied by researchers (Ahn *et al* 2007). The performance of tubular heat exchanger operating under wet surface conditions investigated by

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researchers (Liu *et al* 2009) and studied the condensate retention and the attendant thermal-hydraulic effect associated with changes in air-side surface wettability. Falling film dry out models and heat and mass transfer problems on various configurations and geometries have been studied by various researchers (Bo Jiao *et al* 2009; Jaroslaw *et al* 2009; Volle, F. *et al* 2009; Roxana *et al* 2010;). Correlations in terms of Nusselt number, equivalent Reynolds number, Prandtl number, corrugation pitch and depth, and inside diameter, for the evaporation heat transfer coefficient and two-phase friction factor of R-134a flowing through horizontal corrugated tubes are proposed by investigators (Suriyan Laohalertdecha *et al* 2011). Effects of inlet restriction on flow boiling instability in a single horizontal microtube were investigated experimentally by investigators (YanFeng Fan *et al* 2012) and concluded that reducing the orifice to the microtube area ratio can increase the heat flux of the onset of flow instability or delay the onset of flow instability. Flow and mass transfer characteristics in axisymmetric sinusoidal wavy-walled tubes with different dimensions were investigated experimentally by the researchers (Yongning Bian *et al* 2013).

Fig.1 Photographic view of a row of horizontal tubes used in the experiments with cooling water droplets falling off the tubes

Correlations for mass transfer coefficient and evaporative effectiveness with Dimensionless enthalpy potential, Reynolds number of cooling water and Reynolds number of air derived by using multiple regression analysis on experimental data on evaporative tubular heat exchanger, which works on the combined principle of cooling tower and shell & tube heat exchangers. The motivation behind the present work is to find out the required solution and to study the mass transfer coefficient and evaporative effectiveness of a row of horizontal tubes [Fig.1], on an evaporative tubular heat exchanger with the variation of Relative humidity and other operating parameters.

Test Facility

The schematic of the experimental test rig fabricated for the present investigation is shown in Fig.2.

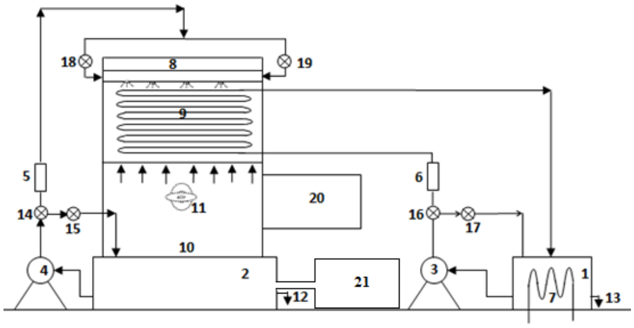

Fig.2 Schematic of evaporative tubular heat exchanger test rig

1: Hot water reservoir. 2: Cooling water reservoir. 3: Hot water Supply Pump.
4: Cooling water supply pump. 5-6: Digital flow meters. 7: Heating elements. 8: Cooling water spray pipe.
9: Test section with a row of tubes. 10: Air duct. 11: Axial flow blower. 12-13: Drains.
14-19: Flow control valves. 20: Humidifier. 21: Feeder tank

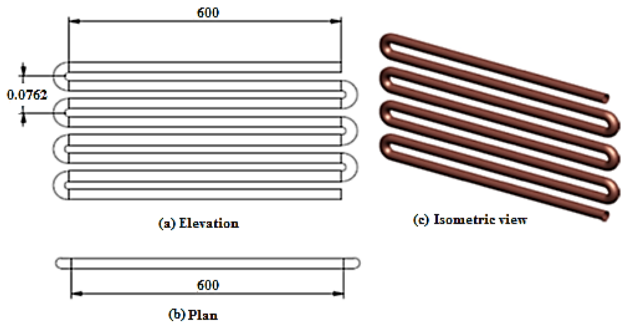

Fig.3 Schematic of a row of horizontal tubes used in the experiments; (a) Elevation, (b) Plan and (c) Isometric view

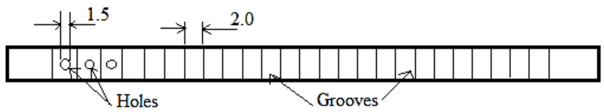

Fig.4 Schematic of spray pipe

A row of eight copper tubes in a single plane is used as shown in the Fig. 3. The inner and outer diameters of the tube used are 0.0234 m and 0.0254 m, respectively. Each tube is inter-connected with the help of U-shaped copper bend of same wall thickness of 0.001 m. Although the bends are not being considered for the calculations for heat transfer analysis, the effective lengths of the copper tubes are considered for calculations of heat transfer analysis and the horizontal projection of each copper tube in a row is 0.6 m. Thus, the active length (l) of a row of tubes is 4.8 m. Area of cross-section of the test section at the top (A_t) is 0.396 m^2 and the area of cross-section of the test section at the test tube level (A_e) is 0.348 m^2. The arrangement of cooling water spray system is fitted over the test section to fall water droplets exactly over a row of tubes. The inner and outer diameters of the spray pipe are 0.0265 m and

0.034 m, respectively. The grooves over the spray pipe are cut whose depth is half the thickness of the spray pipe thickness. The pitch of the grooves is 0.002 m and the holes are drilled in each of the grooves. The diameter of each the hole is 0.0015 m. The schematic of spray pipe is shown in Fig. 4. The temperatures of the hot water at inlet and at outlet of the coil of eight tubes are measured with the help of temperature sensors, the tube surface temperature at the middle of the coil is measured with the help of calibrated chip sensor and counter checked by pre-calibrated infrared thermometer, having resolution of 0.1^oC and the percentage of error in accuracy of ±1.5%. The temperature sensors are then connected to the 32-channel programmable data logger, having resolution of 0.1^oC, with the help of which the temperatures at various points are taken in the form of tables directly. The data logger is connected to the computer with the help of serial port. The software named as the 'process analyser' is used to get the observations online. The flow rates of cooling water and hot water are measured with the help of calibrated flow meters. Two turbine flow meters with digital display are used separately to measure the flow rates of cooling water as well as of hot water. The resolution of flow meters used is 0.1 litres per minute. The velocity of the air leaving the test section at the top is measured at different points of test unit to find the average velocity, with the help of pre-calibrated digital anemometer. The range, resolution and accuracy of the digital anemometer used are 0.1 to 30.0 m/s, 0.1 m/s and ±2%, respectively. Relative humidity is varied at inlet of the test unit with the help of humidifier in the range of 50% to 90% and is measured with help of digital hand held Hygro-thermometer with a range, resolution and accuracy from 0% to 100%, 0.1% and ±2%, respectively and the temperature range from 10^oC to 60^oC.

Experimental Procedure

Firstly the cold was made to flow over a row tubes for about 50 hours until the tubes were fouled, so as to match the actual conditions. For the first set of readings, the air velocity at the top of the test section was fixed at 0.8 m/s, mass flow rate of hot water was fixed at 3.00 x 10^{-2} kg/s, mass flow rate of cooling water was fixed at 2.00 x 10^{-2} kg/s and the Relative humidity at the inlet of test unit varied from 50% to 90% with a step increase of 10%. For the second, third, fourth and fifth sets of readings, first set was repeated except that the mass flow rate of cooling water was varied from 2.00x10^{-2} kg/s to 6.80x10^{-2} kg/s with step increase of 1.20x10^{-2} kg/s and the temperatures of hot water at inlet & outlet of the coil, the temperature of cooling water at the above & below the test unit, average temperature of the tube surface, dry bulb and wet bulb temperatures of air at the inlet & outlet of test unit were noted. The above sets of experiments were repeated at air velocities at the top of test section as 1.6 m/s, 2.4 m/s, 3.2 m/s and 4.0 m/s. Each set of observations contained 5 observations corresponding to Relative humidity as 50%, 60%, 70%, 80% and 90%, thus total sets of observations taken were 125. Beside this, observations were also taken without the air flow keeping mass flow rate of hot water fixed at 3.00 x 10^{-2} kg/s and varying mass flow rate of cooling water from 2.00 x 10^{-2} kg/s to 6.80 x 10^{-2} kg/s.

Range of Operating Variables

Mass flow rate of cooling water in the experiments was taken in the range of 2.00 x 10^{-2} kg/s to 6.80 x 10^{-2} kg/s (Temperature $28±1^oC$) and the flow rate of hot water was taken as 3.00 x 10^{-2} kg/s (Temperature $55±1^oC$). The temperature of the air was taken as $27±1^oC$. Reynolds Number of cooling water was taken in the range of 10.26 to 37.14; Reynolds Number for air was taken in the range of 1245.99 to 6416.95. Relative humidity varied from 50% to 90% with a step increase of 10%.

Governing Equations

The heat dissipation rate from a row of tubes, when both air and cooling water flowing simultaneously can be written as:

$$Q_{wa} = W_h C_p (t_{h1} - t_{h2}) \quad (1)$$

Reynolds number of air and cooling water are determined as:

$$\mathrm{Re}_a = \frac{\rho_a V_t D_o}{\mu_a} \quad (2)$$

$$\mathrm{Re}_w = \frac{4\Gamma}{\mu_w} \quad (3)$$

Liquid film flow rate per unit length of cooling water, Γ is given as:

$$\Gamma = \frac{W_w}{2l} \quad (4)$$

where, l is the active length of a row of tubes and is taken as 4.8 m and W_w is the mass flow rate of cooling water.
Mass transfer coefficient is calculated from:

$$K = \frac{Q_{wa}}{A_o (i_{s,tc} - i_a)} \quad (5)$$

where, $(i_{s,tc} - i_a)$ is the enthalpy potential, the difference of enthalpy of saturated air at the average tube surface temperature and the enthalpy of air at the inlet of heat exchanger. A_o is the outside surface area of the tube and is calculated as 0.383 m^2.
The evaporative effectiveness can be expressed as:

$$EE = \frac{Q_{wa}}{Q_w} \quad (6)$$

where, Q_w is the heat dissipation rate from a row of tubes, when only cooling water flowing at same operating conditions and is calculated as:

$$Q_w = W_h C_p (t_{h1} - t_{h2}) \quad (7)$$

Correlations

Correlations for mass transfer coefficient and evaporative effectiveness with dimensionless enthalpy potential, Reynolds number of cooling water and Reynolds number of air, derived by using multiple regression analysis and their qualitative effects are studied and are depicted in Figs. [5-14]. It is found from Figs. [5-9], that the mass transfer coefficient increase with the Reynolds number of cooling water, because of the fact that for higher cooling rate due to evaporation, higher flow rate of cooling water is required. Mass transfer coefficient is also found to be increased with Reynolds number of cooling water as the mass transfer coefficient depends on the amount of water evaporated. Quantitatively, K gets enhanced by 13.80% to 88.28% at 50% Relative humidity, 11.76% to 89.93% at 60% Relative humidity, 16.08% to 93.97% at 70% Relative humidity, 17.30% to 126.90% at 80% Relative humidity, 17.54% to 110.20% at 90% Relative humidity with respect to the value corresponding to the initial value with Reynolds number of cooling water.

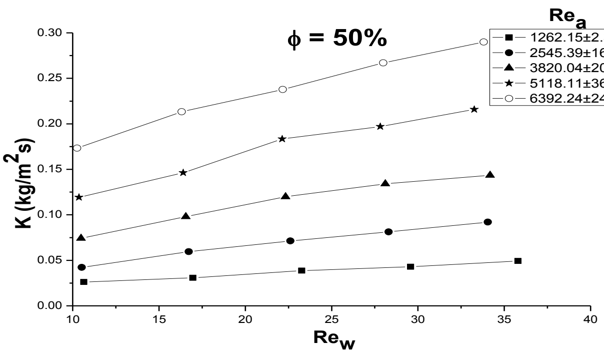

Fig.5 Effect of film Reynolds number of cooling water on mass transfer coefficient at 50% Relative humidity

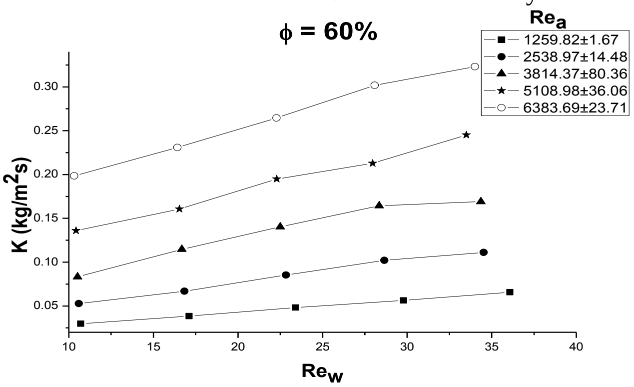

Fig.6 Effect of film Reynolds number of cooling water on mass transfer coefficient at 60% Relative humidity

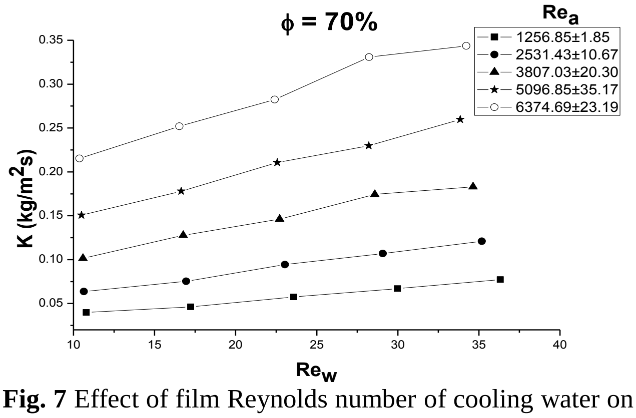

Fig. 7 Effect of film Reynolds number of cooling water on mass transfer coefficient at 70% Relative humidity

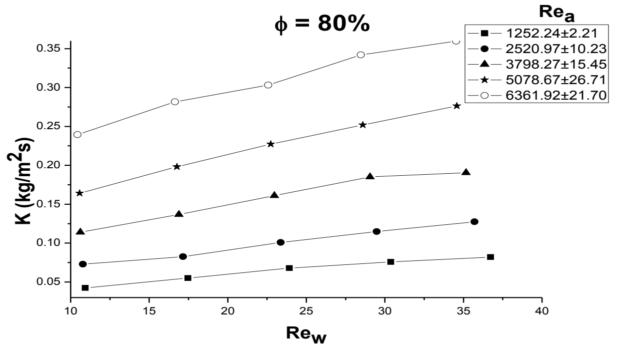

Fig. 8 Effect of film Reynolds number of cooling water on mass transfer coefficient at 80% Relative humidity

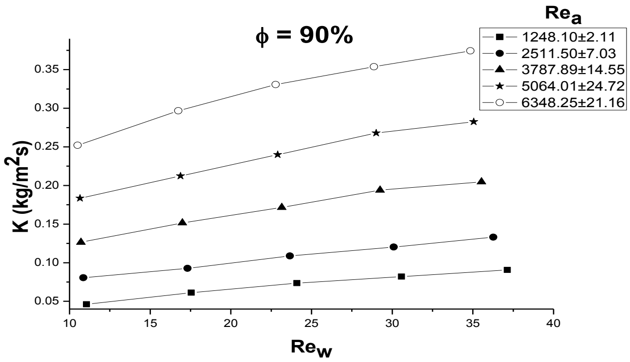

Fig. 9 Effect of film Reynolds number of cooling water on mass transfer coefficient at 90% Relative humidity

The qualitative effects of dimensionless enthalpy potential on evaporative effectiveness with specified values of humidity are shown in Figs. [10-14]. It is observed that evaporative effectiveness get enhanced with the dimensionless enthalpy potential at specified values of Reynolds number of cooling water and Relative humidity because of the fact that evaporative heat transfer increases at higher flow rate of air. However it can be observed that at lower Reynolds number of cooling water, evaporative effectiveness increases, as the effect of air predominates to enhance the evaporative effect. Quantitatively, EE gets enhanced by 1.80% to 83.30% at Re_a=1255.23±9.24, 7.40% to 246.0% at Re_a=2533.33±28.86, 7.50% to 109.00% at Re_a=3811.06±28.80, 8.70% to 154.00% at Re_a=5096.78±57.49, 1.70% to 31.40% at Re_a=6372.02±44.93 with respect to the value

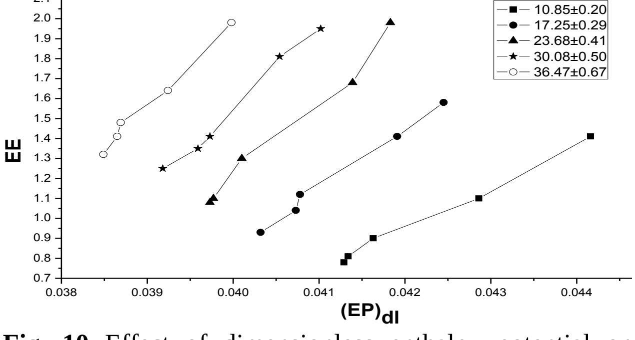

Fig. 10 Effect of dimensionless enthalpy potential on evaporative effectiveness

corresponding to the initial value with dimensionless enthalpy potential.

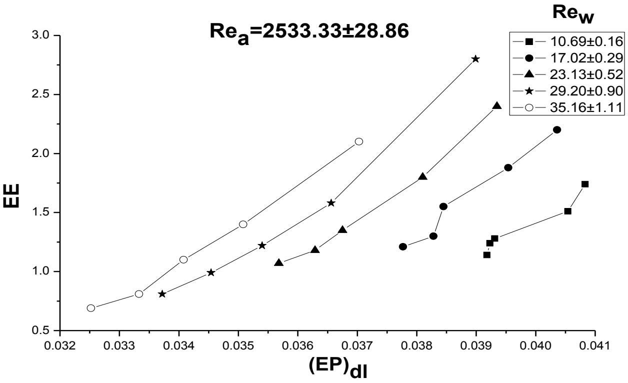

Fig. 11 Effect of dimensionless enthalpy potential on evaporative effectiveness

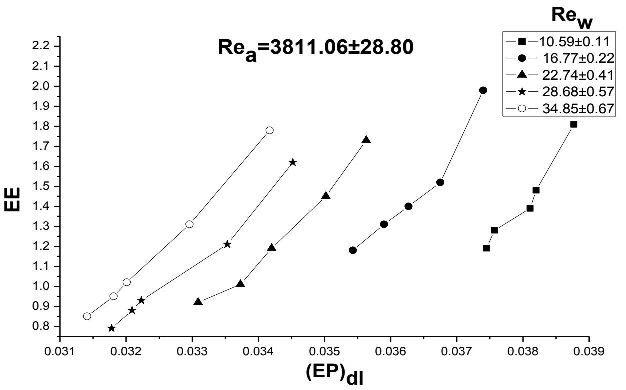

Fig. 12 Effect of dimensionless enthalpy potential on evaporative effectiveness

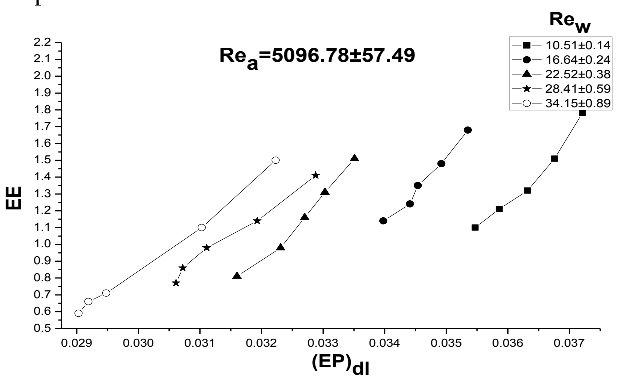

Fig. 13 Effect of dimensionless enthalpy potential on evaporative effectiveness

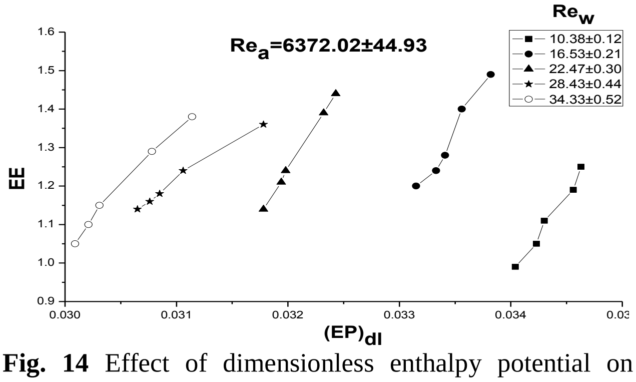

Fig. 14 Effect of dimensionless enthalpy potential on evaporative effectiveness

A multiple regression analysis of experimental data collected for a row of tubes is used to find the correlations and to improve the system; the computed results are used to develop the correlations of mass transfer coefficient and evaporative effectiveness in terms of dimensionless numbers.

Mass transfer coefficient with dimensionless enthalpy potential, Reynolds number of cooling water and Reynolds number of air with the mean and standard deviation of -0.19 and 6.11, respectively are correlated as:

$$K=0.00245(EP)_{dl}^{3.20}(Re_w)^{0.77}(Re_a)^{1.51} \tag{8}$$

(For $1245.99 \le Re_a \le 6376.99; 10.32 \le Re_w \le 17.55; 0.033 \le (EP)_{dl} \le 0.044$)

Mass transfer coefficient with dimensionless enthalpy potential, Reynolds number of cooling water and Reynolds number of air with the mean and standard deviation of -0.06 and 4.81, respectively are correlated as:

$$K=0.00604(EP)_{dl}^{2.65}(Re_w)^{0.82} (Re_a)^{1.44} \tag{9}$$

(For $1249.11 \le Re_a \le 6408.36; 22.14 \le Re_w \le 30.58; 0.030 \le (EP)_{dl} \le 0.041$)

Above correlations show a good agreement between experimental and predicted values of mass transfer coefficient with an error of ±5%.

Evaporative effectiveness with dimensionless enthalpy potential, Reynolds number of cooling water and Reynolds number of air with the mean and standard deviation of -0.06 and 3.57, respectively are correlated as:

$$EE=0.18(EP)_{dl}^{6.55}(Re_w)^{6.22} (Re_a)^{1.07} \tag{10}$$

(For $1245.99 \le Re_a \le 6367.53; 10.32 \le Re_w \le 11.05; 0.034 \le (EP)_{dl} \le 0.044$)

Evaporative effectiveness with dimensionless enthalpy potential, Reynolds number of cooling water and Reynolds number of air with the mean and standard deviation of -0.04 and 2.87, respectively are correlated as:

$$EE=0.39(EP)_{dl}^{6.77}(Re_w)^{5.29} (Re_a)^{1.07} \tag{11}$$

(For $1248.56 \le Re_a \le 6376.99; 16.32 \le Re_w \le 17.55; 0.033 \le (EP)_{dl} \le 0.042$)

Below correlations show a good agreement between experimental and predicted values of evaporative effectiveness with an error of ±4.0%.

The computed values of mass transfer coefficient are compared with those of the predicted ones, obtained by Eq. (8) and Eq. (9), collectively for whole range of Reynolds number of water and air and are shown in Fig. 15. It is found that 95% of the computed values of K lie with in ±5% of their predicted values.

The computed values of evaporative effectiveness are compared with those of the predicted ones, obtained by Eq. (10) and Eq. (11), collectively for whole range of Reynolds number of water and air and are shown in Fig. 16. It is found that 96% of the computed values of EE lie with in ±4% of their predicted values.

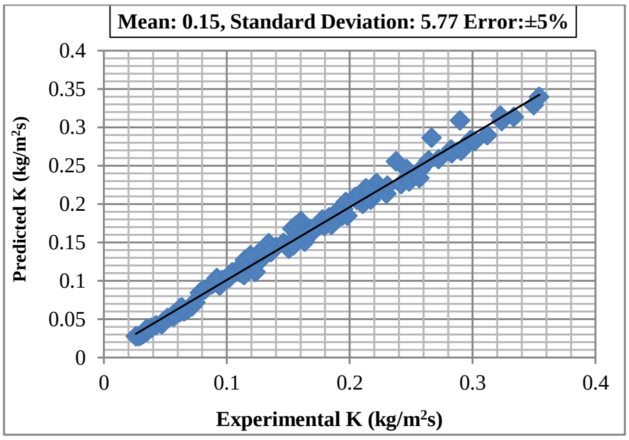

Fig. 15 Comparison between experimental and predicted mass transfer coefficient for full range of Reynolds number of water and air

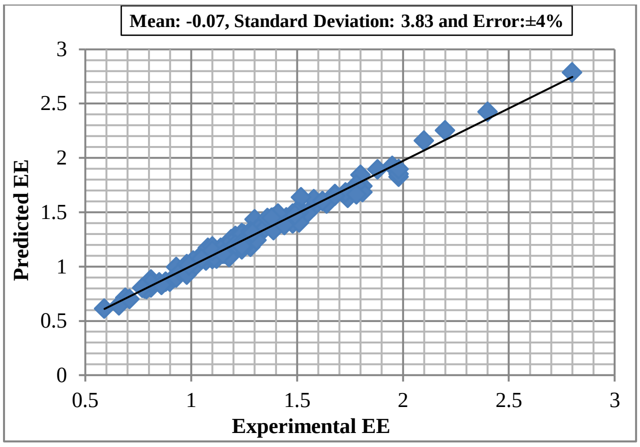

Fig. 16 Comparison between experimental and predicted evaporative effectiveness for full range of Reynolds number of water and air

Conclusions

The cooling water, homogeneously distributed along the tubes of the coil and falling freely from top most tube to the bottom most tube and air was supplied from underneath the test unit with a controlled amount of Relative humidity, was studied experimentally at steady state conditions. The correlations for mass transfer coefficient and evaporative effectiveness developed for general purposes indicate that as the cooling water film flow rate increases while air flow rate is kept constant, mass transfer coefficient and evaporative effectiveness increases. However the increase is relatively less with the change in air flow rate at fixed values of cooling water film Reynolds number.

For the ranges of flow rate and Relative humidity chosen, the correlations presented may prove to be useful in the design of an evaporative tubular heat exchanger, where minimum wetting rate must be known to determine the minimum liquid recycling ratio for an evaporative tubular heat exchanger. So, it can be concluded that for high Relative humidity the Reynolds number of air must be suitable enough for better cooling effects, as it is observed from the data collected through experiments that for low Reynolds number of air, evaporative effectiveness slightly increased, then upon increasing Reynolds number of air, evaporative effectiveness greatly increased and upon further increasing Reynolds number of air, evaporative effectiveness slightly increased. Based on the experimental data, Eq. (8), Eq. (9), Eq. (10) and Eq. (11) have been developed that may be used to calculate mass transfer coefficient and evaporative effectiveness for better designing of an evaporative tubular heat exchanger.

Nomenclature

A: Area, m^2
C_p: Specific heat of water at constant pressure, J/kgK
D: Diameter, m
EE: Evaporative effectiveness
i_a: Enthalpy of air, J/kg
$i_{s,tc}$: Enthalpy of saturated air at average wall temperature, J/kg
i_{fg}: Latent heat of vaporization of water at inlet temperature, J/kg
K: Mass transfer coefficient, kg/m^2s
Q: Heat flow rate from the tube, W
Re: Reynolds number
t: Temperature, ^oC
V: Velocity of air, m/s
W: Mass flow rate, kg/s

Greek Symbols

Γ: Liquid film flow rate, kg/sm
μ: Dynamic viscosity, Ns/m^2
ρ: Density, kg/m^3
φ: Relative humidity

Subscripts

a: Air
e: Projection on horizontal plane
h: Hot water
o: Outside
s: Saturated
t: Top of the test section
w: Cooling water
wa: water & air
1: Inlet
2: Outlet

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