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Research Article

Experimental Study of Falling Film Heat Transfer on a Horizontal Tube of an Evaporative Tubular Heat Dissipator

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Abstract

The present article deals with the experimental investigation of onset and permanent dry out heat flux and mass transfer coefficient at which dry patch formation just starts and permanent condition of dry patch takes place over the surface of a tube of evaporative tubular heat dissipator. A single horizontal tube of evaporative tubular heat dissipator through which hot fluid is flowing is subjected to simultaneous flows of water from the top and air flow from underneath the test unit, with controlled amount of moisture. The primary objective of the present study is to derive the correlations of onset and permanent dry out heat flux and mass transfer coefficients in terms of dimensionless numbers, using the multiple regression analysis of the experimental observations at a wide range of operating variables. The present empirical results show good agreement with the experimental results. Derived correlations from experimental data show that as the film flow rate increases, heat flux needed to cause a permanent or temporary dry patch formation increases provided that the air flow rate is constant. Mass transfer coefficients pertaining to onset and permanent dry patches are also estimated over a wide range of operating conditions. Correlations derived are helpful in improvement of the design of heat transfer devices and many other engineering applications.

Keywords: *Horizontal Tube, Heat Dissipator, Heat Transfer*

Introduction

The rate of heat transfer that takes place between falling liquid film and a horizontal tube of an evaporative tubular heat dissipator, have been studied experimentally, Falling liquid film is widely used in refrigeration systems, petroleum refineries, power plants and chemical industries and utilized in dairy and food industries. The different types of evaporators like forced circulation type, natural circulation type, falling/rising film tubular type, plate evaporative type and multiple effect evaporative type, and different variations in processing techniques have been developed in the past to take into account of different product characteristics and operating parameters. Evaporative tubular heat dissipator 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 dissipator 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 dissipator. A lot of research has been carried out to investigate the heat transfer coefficient over the last few decades (Hartley DE *et al*, 1964; Zuber, N *et al*, 1966; Hodgson JW *et al*, 1968; Simon FF *et al*, 1970; Munakata, T *et al*, 1975; Bankoff, SG *et al*, 1978.) to enhance the performance of evaporative heat dissipation. In evaporative cooling, three flow modes can be observed namely droplet, jet or column and sheet or film modes. These flow patterns are described by researchers (Armbruster *et al*, 1998) and the drop wise evaporation on hot surfaces experimentally studied by researchers (Banacina *et al*, 1979). The mechanism of water film formation over a horizontal tube was investigated and the experiments with sub cooled water film over an electrically heated horizontal cylinder were conducted (Ganic *et al*, 1980). The combinations of film flow rate and heat flux at which film breakdown occurs were determined & observed that the heat flux which is required to cause a dry patch increases with film flow rate and found that there are two types of flow modes associated with liquid films. The first is the droplet mode, which is related to lower flow rates and the second is the jet mode which is related to higher flow rates. Study of heat and mass transfer processes for a vertical tube evaporative cooler carried out later (Perez- Blanco, H *et al*, 1984). The correlations of dry-out mass transfer coefficients for a horizontal and vertical row of tubes, respectively on an evaporative heat dissipator without

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taking Relative humidity into account were developed (Kumar, R *et al*, 1999). Simplified model for indirect contact evaporative cooling tower behaviour was given (Pascal Stabat *et al*, 2003). Functional or Operator representation of numerical heat and mass transport models and a numerical computational procedure was described to determine a multidimensional functional or an operator for the representation of the computational results of a numerical transport code (Danko, D *et al*, 2006). Adiabatic film cooling effectiveness of different patterns measured using heat and mass transfer analogy method was determined (Yuzhen Lin *et al*, 2006), as an advanced cooling scheme to meet increasingly stringent combustor cooling requirements, multi-hole film cooling has received considerable attention. Performance of tubular heat exchanger operating under wet surface conditions were investigated (Liu, L *et al*, 2009). The liquid film dry out model for predicting critical heat flux in annular two phase flow was discussed (Bo Jiao *et al*, 2009). A simple dissipation model of circular hydraulic jump (Jaroslaw Mikielewicz *et al*, 2009) and the phenomenon of hydraulic jump formed as a result of circular jet impingement on the horizontal surface were discussed. Similar studies regarding evaporative heat dissipation has been reported by various other researchers (Fujita, T *et al*, 1978; Arefyev, KM *et al*, 1979; Grissom, W *et al*, 1981; Hallett, VA, 1996; Volle, F *et al*, 2009; Roxana Grigore *et al*, 2010; Muhammad, M *et al*, 2011).

Experimental data on dry patches for evaporative tubular heat dissipator which works on the principle of cooling tower and heat exchangers is helpful in designing the more efficient evaporative heat dissipator. So the present study of the investigations has been conducted to find out the combination of cooling water film rates, air flow rates, heat flux and mass transfer coefficient at which dry patches just occur and then permanently stay on the surface of the tube. The effects of operating parameters like Reynolds number of air, Reynolds number of cooling water and dimensionless enthalpy potential of air on the film breakdown are also studied.

Experimental Test Rig

The schematic of experimental test rig fabricated for the present investigation is shown in Fig.1.

The U-shape of the single horizontal copper tube in test unit is used to ensure the fully developed flow in the tube. The inner and outer diameters of the tube used are 0.0234 m and 0.0254 m, respectively. Active length (l) of the tube is 0.6 m so; the horizontal projection considered for the tube is 0.6 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 temperatures of the hot water at inlet and at outlet of a tube are measured with the help of RTD 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^{o}C and the percentage of error in accuracy of ±1.5%. The temperature

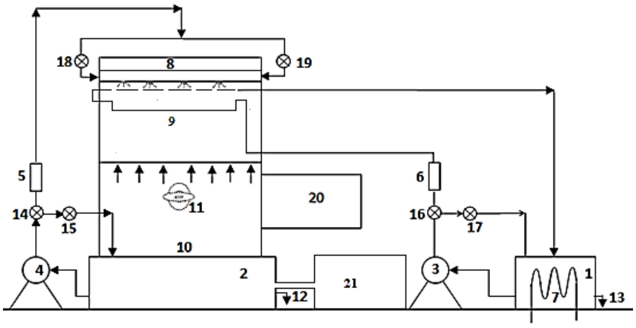

Fig.1 Schematic of evaporative tubular heat dissipator test rig

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

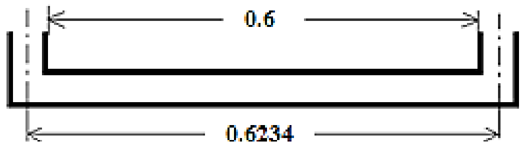

Fig.2 Schematic of copper tube used in the experiment

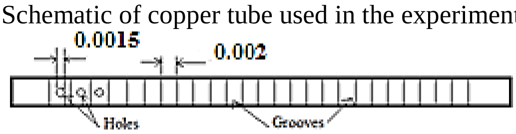

Fig.3 Schematic of spray pipe

sensors are then connected to the 32-channel programmable data logger, having resolution of 0.1^{o}C, 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 percentages of error in accuracy for both the flow meters are ±5% and ±2% for minimum and maximum flow rates, respectively. The resolution of flow meters is 0.1 litres per minute. The velocity of the air leaving the test section from the top is measured at different points at the top 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. The dry patches are observed with the help of cameras and screen. Relative humidity at inlet and at outlet of the test unit 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^{o}C to 60^{o}C.

Experimental Procedure

The cold was made to flow over a tube for about 50 hours until the tube was fouled, so as to match the actual condi-

-tions. For the first set of readings, the air velocity at the top of the test section was fixed at 0.8 m/s and the relative humidity at the inlet of test unit varied from 50% to 90% with a step increase of 10%. The flow rate of cooling water initially at 1.70×10^{-2} kg/s to 5.10×10^{-2} kg/s with step increase of 0.85×10^{-2} kg/s, the initial value of flow rate was fixed more than the minimum wetting rate, which is defined as the rate of cooling water at which the unit length of the copper tube becomes completely wet at no heat load and the hot water then made to flow through the tube at 60 ± 1^{o}C to provide the heat load and the heat flux was increased by increasing the flow rate of hot water in small steps and the steady state was achieved. The heat load was increased till the dry patches appeared for the first time on the outer surface of the tube and the temperature of hot water at the inlet and at the outlet of tube was noted, also the temperature of cooling water at the inlet of spray pipe, average temperature of the tube surface, dry bulb and wet bulb temperature of air at the inlet of test section was noted. This condition of formation of dry patch is referred to the condition of onset dry patch formation and corresponding heat load is termed as the onset dry out heat flux. Most of the dry patches were rewetted by the cooling water but some of the dry patches remained dry. These dry patches were rewetted manually by putting more drops of cooling water manually by dropper over the surface of tube. After getting disappearance of the dry patches from the outer surface of the tube, the heat load was again increased by increasing the flow rate of hot water in small steps and the steady flow condition was achieved. This process was continued till the state came that the dry patches were not disappearing even after increasing the flow rate of the cooling water or by placing more droplets manually over the surface of tube. This condition of formation of dry patch is referred to the condition of permanent dry patch formation and corresponding heat load is termed as the permanent dry out heat flux. At this state, the temperatures of hot water at the inlet and at the outlet of tube were noted; also the temperature of cooling water at the inlet of spray pipe, average temperature of the tube surface, dry bulb and wet bulb temperatures of air at the inlet of test section were noted. The experiment was repeated for second, third, fourth and fifth set of readings 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 at Relative humidity 50%. Similarly, the set of observations recorded at air velocities at the top of test section as 0.8 m/s, 1.6 m/s, 2.4 m/s, 3.2 m/s and 4.0 m/s at Relative humidity 60%, 70%, 80% and 90% to get 25 sets of observations in total. Each set of observations contain 5 observations.

Range of Operating Variables

The flow rate of process fluid in the experiment is taken in the range of 2.00×10^{-2} kg/s to 11.00×10^{-2} kg/s and the flow rate of cooling water is taken in the range of 1.70×10^{-2} kg/s to 5.10×10^{-2} kg/s. The temperature of the process fluid in the experiment is taken as 60 ± 1^{o}C and the temperature of the cooling water is taken as 28 ± 1^{o}C. The temperature of the air is taken as 27 ± 1^{o}C and the velocity of the air is controlled in the range of 0.8 m/s to 4.0 m/s, with controlled amount of moisture. Reynolds Number of cooling water is taken in the range of 71.42 to 223.90 and the Reynolds Number for air is taken in the range of 1258.33 to 6297.25. Relative humidity varied from 50% to 90%.

Governing Equations

The heat dissipation rate from a tube can be written as:

$$Q_{wa} = W_p C_p (t_{h1} - t_{h2}) \tag{1}$$

Reynolds number of air and cooling water are determined as:

$$Re_{,a} = \rho \, V_t \, D_o / \, \mu_a \tag{2}$$

$$Re_{,w} = 4\Gamma / \, \mu_w \tag{3}$$

Mass flow rate of cooling water, Γ is found as:

$$\Gamma = W_w / \, 2l \tag{4}$$

where, l is the active length of tube and its value used here is 0.6 m.

Dry out heat flux which causes the onset and permanent dry patch is determined by using the following relation:

$$q_{on} \text{ or } q_p = Q_w / A_o \tag{5}$$

where, outside surface area of the test tube (A_o) is calculated as 0.376 m^2.

Mass transfer coefficient for onset and permanent dry patch condition is calculated from:

$$K_{on} \text{ or } K_p = Q_{wa} / A_o (i_{s,tc} - i_a) \tag{6}$$

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 dissipator.

The dimensionless enthalpy potential can be expressed as:

$$(EP)_{dl} = (i_{s,tc} - i_a) / i_{fg} \tag{7}$$

where, i_{fg} is the latent heat of vaporization of water at inlet temperature

Onset and Permanent Correlations

The effects of Reynolds number of water on the onset and permanent dry out heat fluxes and mass transfer coefficients for some selected values of Reynolds number of air are shown in Figs. [4-7]. It is found from these figures that the values of onset and permanent dry out heat flux, onset and permanent mass transfer coefficients increase with Reynolds number of water. It is because of

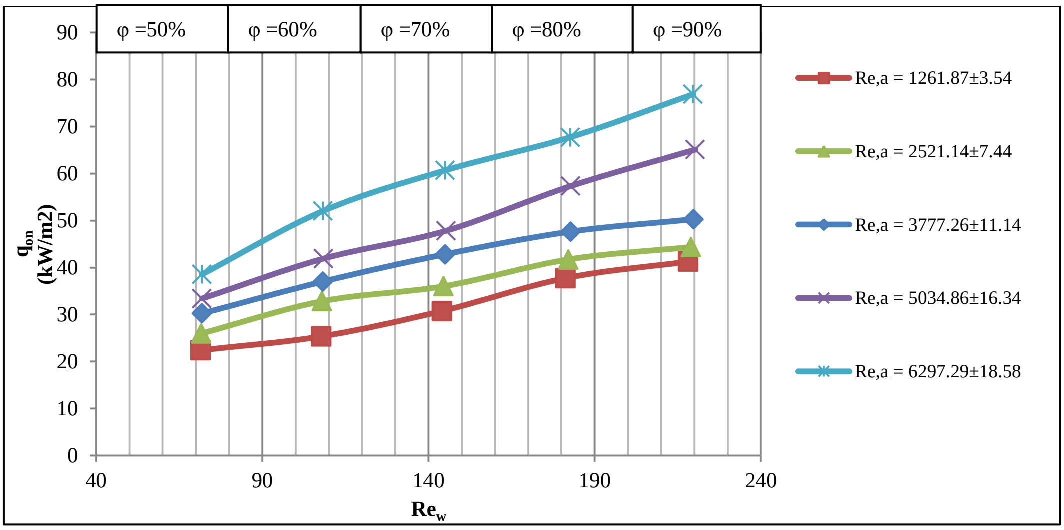

Fig.4 Effect of film Reynolds number of water on the onset dry out heat flux of a single tube

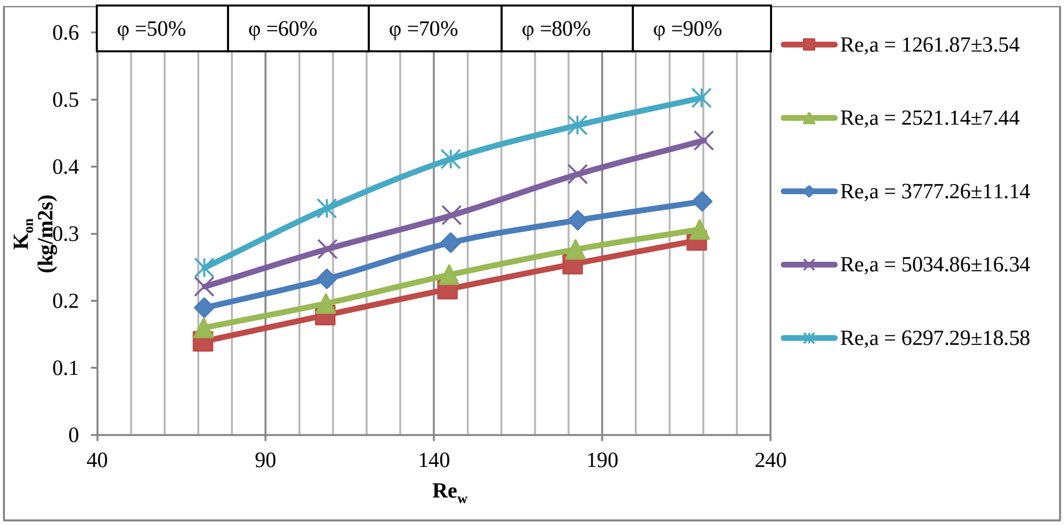

Fig.5 Effect of film Reynolds number of water on the onset dry out mass transfer coefficient of a single tube

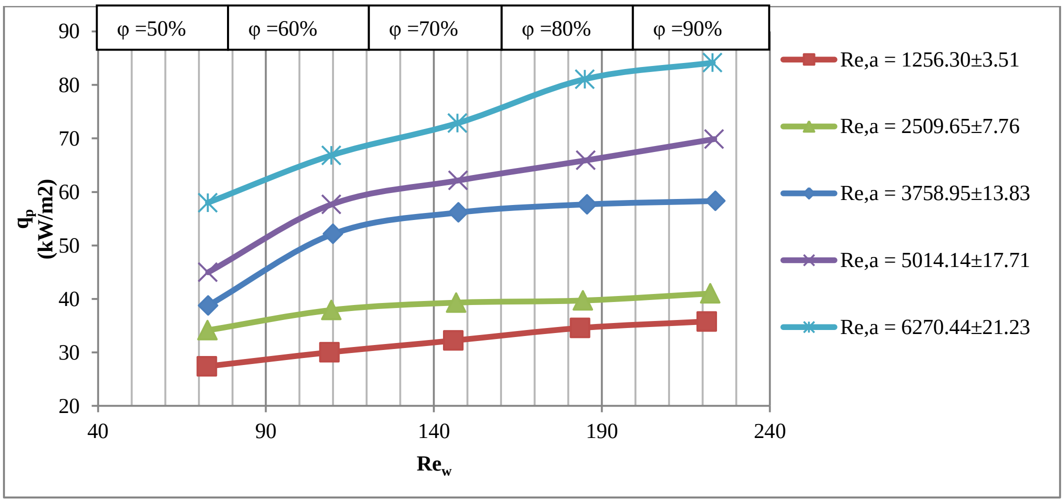

Fig.6 Effect of film Reynolds number of water on permanent dry out heat flux of a single tube

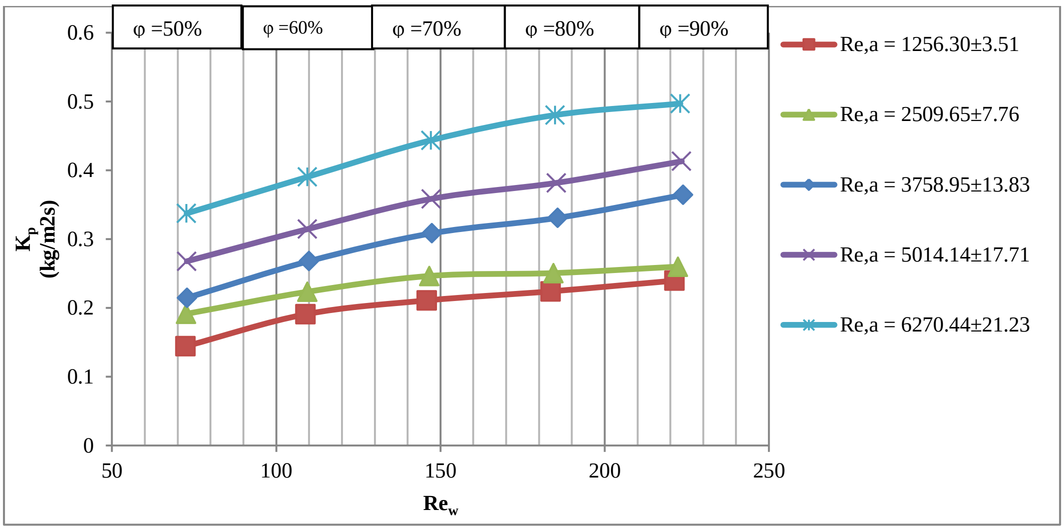

Fig.7 Effect of film Reynolds number of water on permanent dry out mass transfer coefficient of a single tube

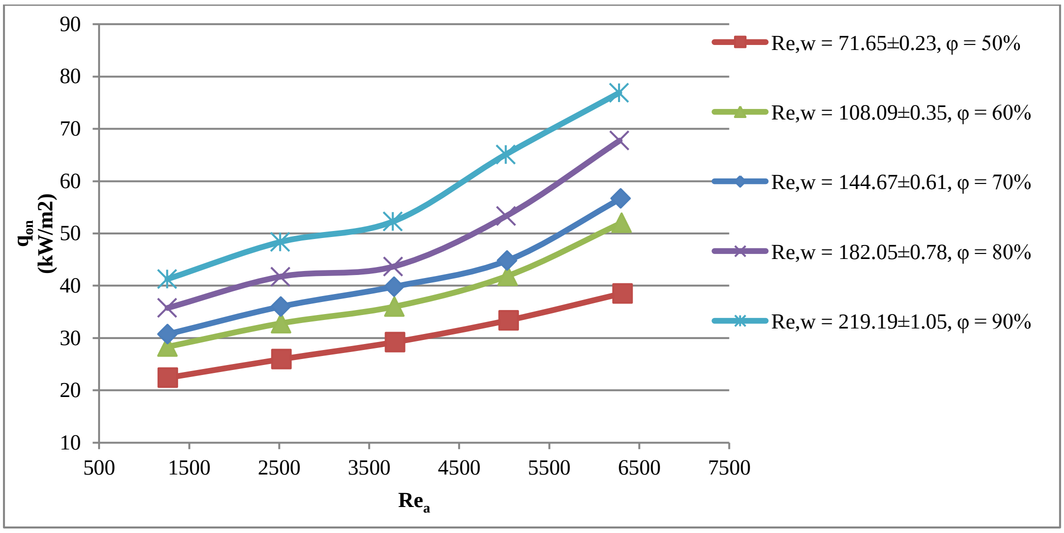

Fig. 8: Effect of film Reynolds number of air on the onset dry out heat flux of a single tube

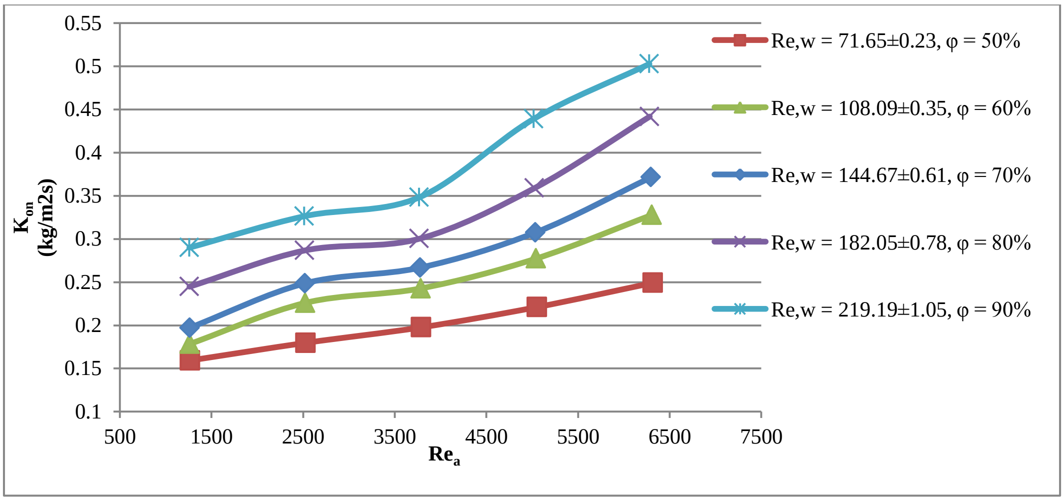

Fig. 9: Effect of film Reynolds number of air on the onset mass transfer coefficient of a single tube

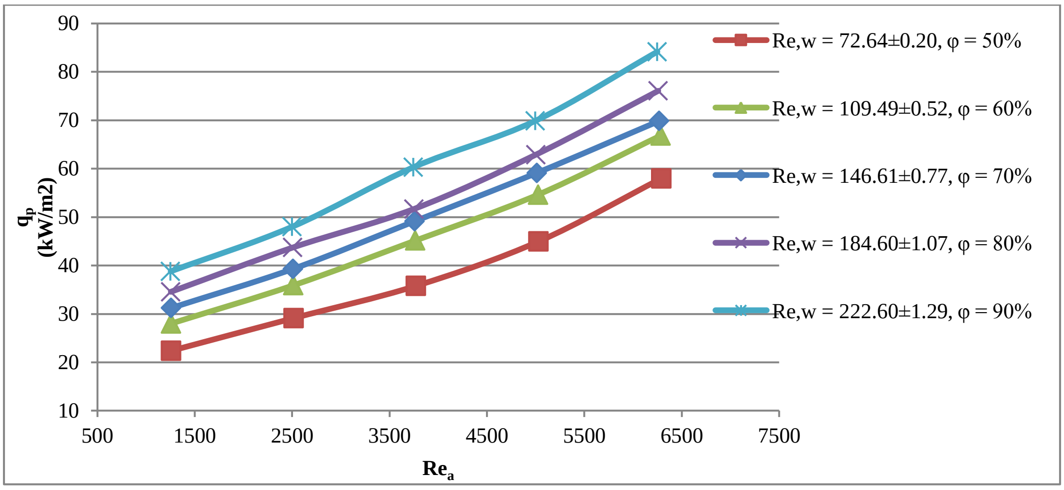

Fig. 10: Effect of film Reynolds number of air on the permanent dry out heat flux of a single tube

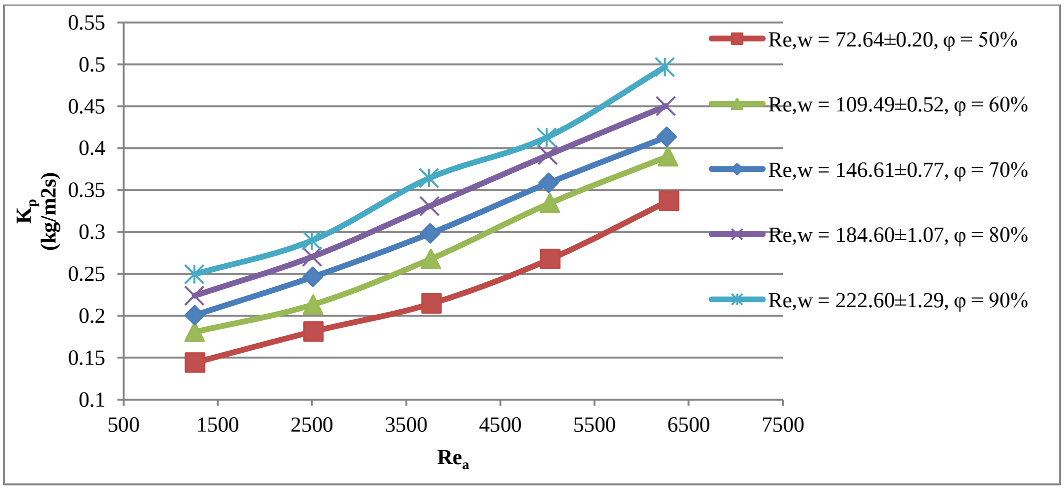

Fig. 11: Effect of film Reynolds number of air on the permanent mass transfer coefficient of a single tube

the fact that higher is the flow rate of cooling water, greater is the heat flux required to evaporate water at faster rate from the tube surface to form the dry patches over the surface of a tube. As mass transfer coefficient is related with the quantity of water evaporated, higher values of mass transfer coefficients are also found at higher values of Reynolds number of cooling water. Quantitatively values of onset dry out heat fluxes for Reynolds number of air 1261.87±3.54, 2521.14±7.44, 3777.26±11.14, 5034.86±16.34 and 6297.29±18.58 get enhanced by 13% to 84%, 26% to 70%, 20% to 43%, 25% 94% and 34% 99%, respectively and values of permanent dry out heat fluxes for Reynolds number of air 1256.30±3.51, 2509.65±7.76, 3758.95±13.83, 5014.14±17.71 and 6270.44±21.23 get enhanced by 9% to 34%, 5% to 20%, 31% to 39%, 28% 50% and 15% to 45%, respectively. Similarly, the quantitative values of onset mass transfer coefficients for Reynolds number of air 1261.87±3.54, 2521.14±7.44, 3777.26±11.14, 5034.86±16.34 and 6297.29±18.58 get enhanced by 12% to 82%, 25% to 70%, 24% to 44%, 25% to 98% and 35% 101%, respectively and values of permanent mass transfer coefficients for Reynolds number of air 1256.30±3.51, 2509.65±7.76, 3758.95±13.83, 5014.14±17.71 and 6270.44±21.23 get enhanced by 9% to 37%, 5% 23%, 20% to 43%, 28% to 54% and 15% to 47%, respectively. In order to achieve the actual situation, a metered quantity of humidity in the test unit is introduced with air entering the test unit.

The results shows that for Re_w = 71.65±0.23 and φ = 50%, q_{on} get increased with Re_a by 15% to 72%, for Re_w = 108.09±0.35 and φ = 60%, q_{on} get increased with Re_a by 29% to 105%, for Re_w = 144.67±0.60 and φ = 70%, q_{on} get increased with Re_a by 29% to 104%, for Re_w = 182.05±0.78 and φ = 80%, q_{on} get increased with Re_a by 8% to 89% and for Re_w = 219.19±1.05 and φ = 90%, q_{on} get increased with Re_a by 7% to 86%, as shown in Fig. 8. Similarly, the variations of K_{on}, q_p and K_p with Re_a are shown in Figs. [9-11], respectively.

It is observed that all the quantities q_{on}, q_p, K_{on} and K_p get enhanced with Re_a, because of the higher value of flow rate of air, both convective effect at the tube wall and the heat flux required to break the cooling water film increase and consequently the dry out mass transfer coefficients

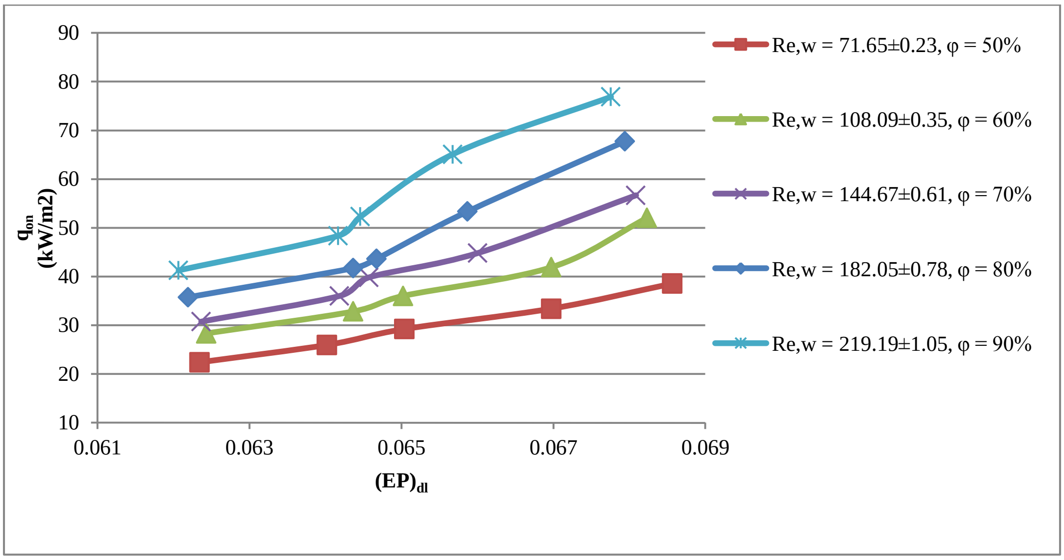

Fig. 12: Effect of dimensionless enthalpy potential on the onset dry out heat flux of a single tube

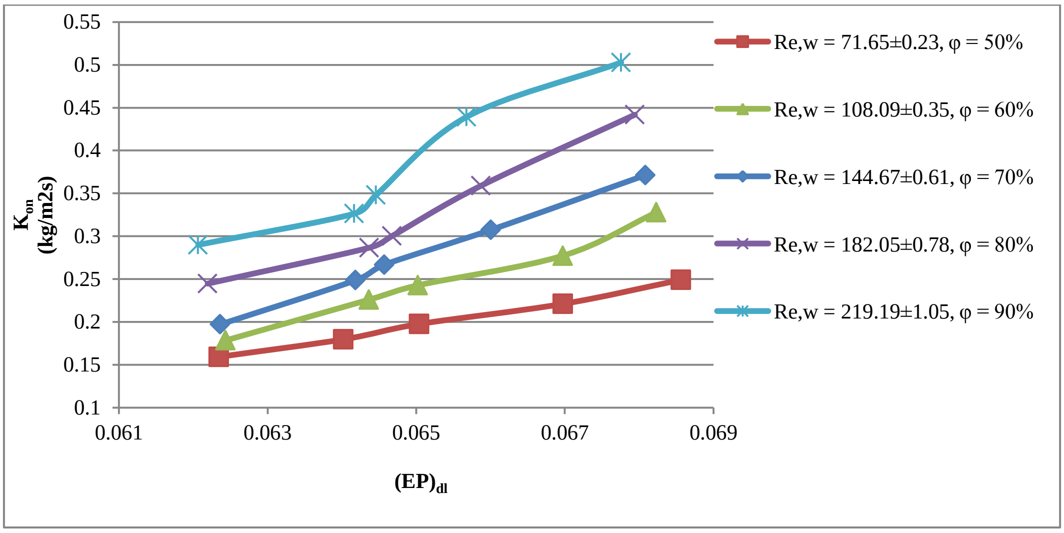

Fig. 13: Effect of dimensionless enthalpy potential on the onset mass transfer coefficient of a single tube

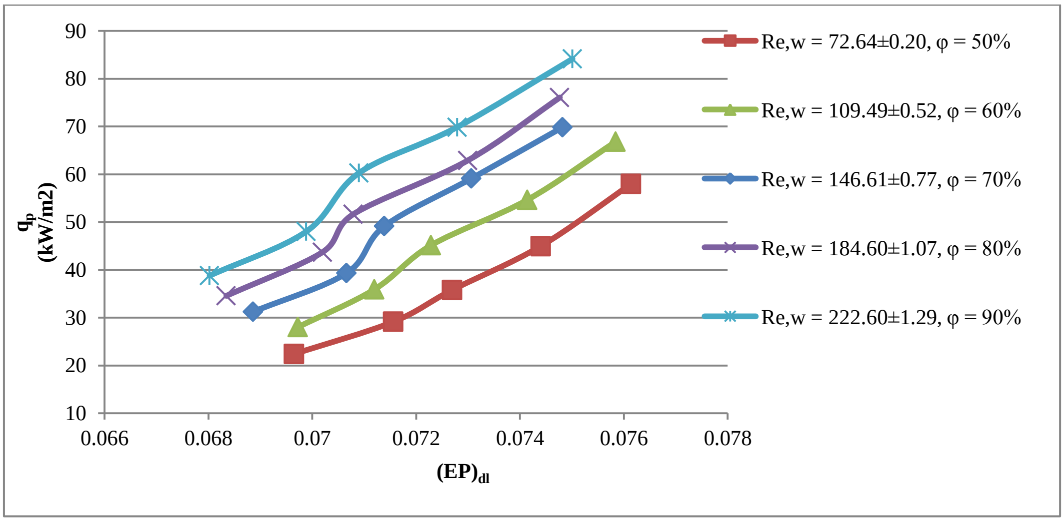

Fig. 14: Effect of dimensionless enthalpy potential on the permanent dry out heat flux of a single tube

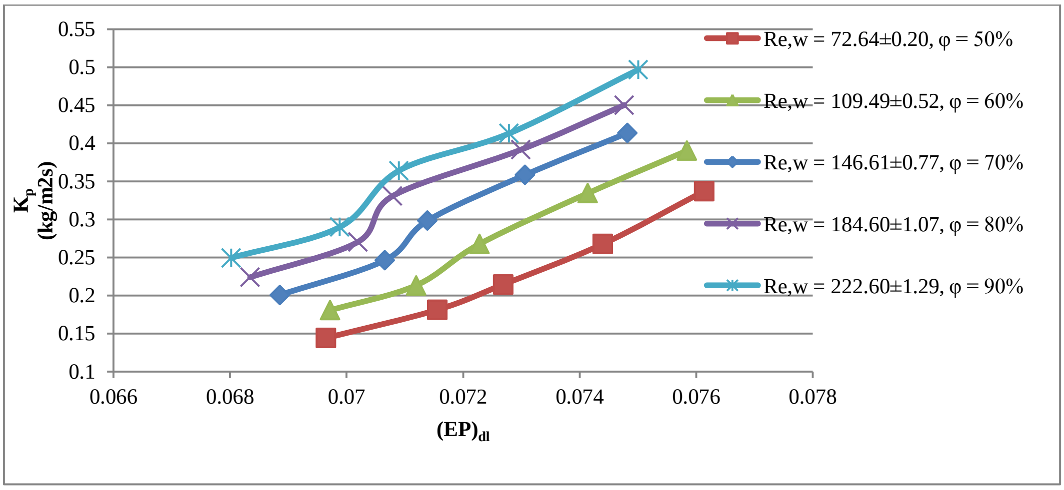

Fig. 15: Effect of dimensionless enthalpy potential on the permanent mass transfer coefficient of a single tube

also increased.

It is observed from Fig. 12, for $Re_w = 71.65\pm0.23$, φ = 50%, $(EP)_{dl}$ with respect to q_{on} increased from 2% to 9%, for $Re_w = 108.09\pm0.35$, φ = 60%, $(EP)_{dl}$ with respect to q_{on} increased from 2% to 8%, for $Re_w = 144.67\pm0.60$, φ = 70%, $(EP)_{dl}$ with respect to q_{on} increased from 2% to 8%, for $Re_w = 182.05\pm0.78$, φ = 80%, $(EP)_{dl}$ with respect to q_{on} increased from 3% to 9% and for $Re_w = 219.19\pm1.05$, φ = 90%, $(EP)_{dl}$ with respect to q_{on} increased from 1% to 7%. Similarly the effects of dimensionless enthalpy potential on q_p, K_{on} and K_p with specified values of Relative humidity and Reynolds number of cooling water are shown in Figs. [13-15] and the quantities q_{on}, q_p, K_{on} and K_p get enhanced with $(EP)_{dl}$. It also can be observed from Figs. [12-15] that at relative humidity 60% and 70%, the variations in q_{on}, q_p, K_{on} and K_p against $(EP)_{dl}$ are abrupt. So it can be estimated from these variations that either the humidity must be low or the velocity of air must be higher enough to increase the cooling effect with humidity.

A multiple regression analysis of the experimental data collected for a tube is used to find the correlations and to improve the system performance in concern with the formation of dry patches; the computed results are used to develop the correlations of onset and permanent dry out heat flux and mass transfer coefficient in terms of dimensionless numbers.

The onset dry out heat flux with Reynolds number of water and air with the mean and standard deviation of -0.41 and 9.47, respectively are correlated as:
For $71.42 \le Re_w \le 220.24$; $1258.34 \le Re_a \le 6315.88$, when hot water inlet temperature was 60±1°C

$$q_{on}= 0.18(Re_w)^{0.48}(Re_a)^{0.37} \quad (8)$$

This correlation shows a good agreement between experimental and predicted values of onset dry out heat flux with an error of ±10%.

The permanent dry out heat flux with Reynolds number of water, with the mean and standard deviation of 1.44 and 7.87, respectively are correlated as:
For $72.44 \le Re_w \le 223.91$; $1252.79 \le Re_a \le 6291.68$, when hot water inlet temperature was 60±1°C

$$q_p = 0.21(Re_w)^{0.48}(Re_a)^{0.50} \quad (9)$$

This correlation shows a good agreement between experimental and predicted values of onset dry out heat flux with an error of ±8%.

The onset mass transfer coefficient with dimensionless enthalpy potential, Reynolds number of water and Reynolds number of air, with the mean and standard deviation as -4.07 and 9.16, respectively are correlated as:
For $0.062 \le (EP)_{dl} \le 0.068$ ($50\%\le\varphi\le90\%$); $71.42 \le Re_w \le 220.24$; $1258.34 \le Re_a \le 6315.88$, when hot water inlet temperature was 60±1°C

$$K_{on} = 0.04\,(EP)_{dl}^{0.18}(Re_w)^{0.49}(Re_a)^{0.004} \quad (10)$$

This correlation shows a good agreement between experimental and predicted values of onset dry out heat flux with an error of ±10%.

The permanent mass transfer coefficient with dimensionless enthalpy potential, Reynolds number of water and Reynolds number of air, with the mean and standard deviation as 0.15 and 7.13, respectively are correlated as:
For $0.068 \le (EP)_{dl} \le 0.076$ ($50\%\le\varphi\le90\%$); $72.44 \le Re_w \le 223.91$; $1252.79 \le Re_a \le 6291.68$, when hot water inlet temperature was 60±1°C

$$K_p = 24.53\,(EP)_{dl}^{3.23}(Re_w)^{0.37}(Re_a)^{0.28} \quad (11)$$

This correlation shows a good agreement between experimental and predicted values of onset dry out heat flux with an error of ±8%.

Conclusions

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 dissipator, where minimum wetting rate must be known to determine the minimum liquid recycling ratio for an evaporative tubular heat dissipator. Mass transfer coefficients pertaining to onset and permanent dry patch conditions increases with cooling water film flow and air flow rates. The effects of variation of humidity is mainly focused in the present work and is observed that onset and permanent dry out heat flux as well as mass transfer coefficient increases slightly initially for lower Relative humidity (50% and 60%) and lower air velocities (0.8 m/s and 1.6 m/s) but for higher Relative humidity (70%, 80% and 90%) and higher air velocities (2.4 m/s, 3.2 m/s and 4.0 m/s), onset and permanent dry out heat flux as well as mass transfer coefficient increases rapidly. So, it can be concluded that for high Relative humidity the Reynolds number of air must be higher for better cooling effects.

Nomenclature

A: Area, m^2
C_p: Specific heat of water at constant temperature, J/kgK
EP: Enthalpy potential
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 flux, W/m^2
Q_{wa}, Q_w: Heat flow rate from the tube and maximum heat flow rate, respectively, W
Re: Reynolds number
RTD: Resistance temperature detector
t: Temperature of hot water, ^oC
V: Velocity of air, m/s
W: Flow rate, kg/s

Greek Symbols

Γ: Mass flow rate, kg/s
Δi: Logarithmic mean enthalpy difference, J/kg
µ: Dynamic viscosity, Ns/m^2
ρ: Density, kg/m^3
φ: Relative humidity

Subscripts

a: Air
dl: Dimensionless
e: Projection on horizontal plane
h: Hot water
o: Outside
on: Onset
p: Permanent
s: Saturated
t: Top of the test section
w: Water
1: Inlet
2: Outlet

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