Experimental study of dryout heat flux and mass transfer coefficient effect on a Single Horizontal Copper tube of an Evaporative Tubular Heat Dissipator

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Abstract- In this investigation, dryout heat fluxes related to the onset and permanent dry patches is to be determined without the air flow in case of single tube. Mass transfer coefficient is determined with water and air flow from the same tube. The total energy dissipated by inside hot fluid when only water is falling is compared with that when both air and water flow past the tube. The ratio of energies dissipated with water and air flow and with only water flow increases with Re_w and Re_a.

Keywords: Single Horizontal Tube, Dry out Heat flux, Mass transfer coefficient, Heat Dissipation

I. INTRODUCTION

Evaporation is one of the most energy intensive processes used in the dairy, food, power plant, chemical industries. It is a unit operation that is used widely in processing foods, chemicals, pharmaceuticals, fruit juices, sugar industries, desalination, dairy products, paper and pulp, and both malt and grain beverages. In this system there is use of both air and water for dissipation of heat. During the evaporators operation two main points are always considered that suitability of the equipment for its best duty and efficient and economical use of the equipment. Therefore, many types of evaporator and many variations in processing techniques have been developed to take into account different product characteristics and operating parameters. The different types of evaporators are, forced circulation evaporators, natural circulation evaporators, falling film tubular evaporators. The performance of this system is depending upon the heat and mass transfer coefficients. Higher the value of these coefficients, the greater would be the effectiveness of the evaporative tubular heat dissipator.

There are very large quantity of water is used in steel plants, petroleum refineries, nuclear power plant, power generation plants, air conditioning and refrigeration industries etc. In all that industries there is continuous supply of cooling water for dissipation of heat energy. It is necessary that the temperature of cooling water should not exceed a certain prescribed value for a particular process plant. Due to expansion of these industries the requirement of cooling water has been increased many times in the past years.

There are many researches in the past have made both analytical as well as experimental studies to enhance the performance of evaporative tubular heat dissipator. The correlations of heat transfer obtained with nucleate boiling of refrigerant in a horizontal tube were studied first by P.L. Dhar, N.J. Dembi, C.P. Arora [1]. Ganic, *et al.* [2], investigated the mechanism of water film formation over a horizontal tube. They conducted experiment with the film is sub cooled and heat transfer from the heated surface is absorbed in the liquid film. The heat transfer coefficient and film break down heat flux data were obtained. Rana , *et al.* [3] made investigation without the use of air flow from a single horizontal tube that is only water is used for heat dissipation and also made experimental with use of both air and water flow for dissipation of heat energy. Parez- Blanco and Bird [4] have carried out studies on the heat and mass transfer processes of vertical tube evaporative cooler. Rana, *et al.* [5] investigated; the mass transfer coefficient with continuous water and air flow has been determined experimentally and also investigated the effect of reynold number of air on mass transfer coefficient. D.B. Murray [6] made comparison of the heat transfer characteristics of staggered and in-line arrays and found that how to increase and decreased the local and overall heat transfer coefficient in each case. Yan and Lin [7] carried out analytical analysis to investigate the result for heat and mass transfer for the system with ethanol film evaporation. X. Hu and A. M. Jacobi [8] conducted experiment by using four working fluids and a range tube diameters, tube spacing and liquid flow rate by changing these parameters the value of reynold number increase or decreased. R. Armbruster [9] conducted experiment under study state condition, the cooling of water by distributed along a horizontal tube and falling freely to another tube below it. Kumar, *et al.* [10-11] finding the correlations of dryout mass transfer coefficients for a horizontal and vertical row of tubes, on an evaporative heat dissipator. Ali S. Alosaimy [12] conducted experiment by using six modes that is sheet, sheet-jet, inline jet, staggered jet, jet-droplet and droplet made comparison between them that when flow become sheet, sheet-jet, inline jet, staggered jet, jet-droplet and droplet. Pascal Stabat and Dominique Marchio [13], presented the simplified model for indirect cooling-tower behavior. The model is devoted to building simulation tools and fulfils several criteria such as simplicity of parametrisation, accuracy, possibility to model the equipment under various operation conditions and short computation time. On the basis of Merkel's theory, the model is described by using the Effectiveness- NTU method. The model introduces only two parameters, air-side and water-side heat-transfer coefficients which can be identified from only two rating points, data easily available in manufacture's catalogues. G. Danko [14] represents the functional or operator for the numerical heat and mass transfer methods to describe the determination of a multi-dimensional functional or an operator for the representation of the computational results of a numerical transport code. The procedure is called numerical transport code functionalization (NTCF). These numerical transport codes are used to solve heat conduction problems in solids. The heat- mass transfer analogy method determined by Yuzhen Lin, *et al.* [15] for the adiabatic film cooling effectiveness of different patterns. This experimental data of cooling scheme are limited in the open literature in terms of different hole pattern and blowing ratio's. Eashwar Serthuraman, *et al.* [16] was studied the mass heat transfer in a rotating smooth curved walled channels. Syed Naveed Ul Hasan [17] conducted experiment for the evaluation of heat transfer coefficient in a vertical tube rising film evaporator (VRF). This experiment was carried out for laminar flow. There was review of the latest findings at describing the heat transfer in horizontal tube falling film evaporator was studied and find by the Paul Schausberger [18]. A.M. Jacobi [19] studies the flow patterns and mode transition for falling-films on flat tubes. Water and Ethylene glycol used as a working fluid. The effect of tube spacing on falling film transition at two different distributing heights was also studied. The effects of protrusion on heat transfer in a micro tube and in a two-dimensional microchannel of finite wall thickness were studied by Muhammad M. Rahman [20]. This work was further studied by the P.K. Das [21] which was the review of critical heat flux during flow boiling through mini and microchannel. The dryout heat flux, mass transfer coefficient and dimensionless enthalpy of evaporative tubular heat Dissipator

was reviwed by Rajneesh [22].

II. AIM OF THE PAPER

The aim of the paper is to find first dryout heat flux which causes onset or permanent dry patch formation on the surface of the copper tube from which process fluid is flowing but without the air flow.

$$q_{on} \text{ or } q_p = Q_w/A_o \qquad (1)$$

Where, Q_w is the heat dissipation rate from the process fluid with only water falling from the top.

The Q_w is mainly depend upon the inlet and outlet of the process fluid when the difference between the inlet and outlet of the process fluid is more then there will be formation of onset dry patches on the surface of the copper tube and the heat dissipation is more and when this difference is small then there is formation of permanent dry patches. This would be depending upon the variation in the mass flow rate of the process fluid.

Now secondly mass transfer coefficient is to find with the flowing of both air and water. The water is fallen from the top and air is flowing underneath the tube.

III. THE EXPERIMENTAL SET-UP

The basic purpose of the experiments was to investigate the effect of Reynolds number of air, process fluid, cooling water and dimensionless enthalpy potential difference this is done on the single tube. The dimension of the acrylic sheet is 0.81 m x 0.6 m. The cooling water pipe is of dimension outer diameter and inner diameter is 0.034 m and 0.0265 m of G.I pipe whose total length was 0.600 m. Holes of 0.0015 m diameter, 0.003 m apart, were drilled along the whole length of the tube. Grooves 0.0015 m deep were provided over the periphery of the tube. Each hole was drilled at the center of the groove, so that the water flowed down without distributing the flow emanating from the neighboring holes. Air is flowing counter-currently to falling water. The inner and outer diameters of copper tube are 0.0254 m and 0.0234 m. The water which falls from the cooling shower is collected in to the cooling water reservoir. There is a screw arrangement in the cooling shower by which the height of the shower can be adjusted for equal distributing of cooling water on the copper tube which contains hot water.

Table: - Description of parts

S.NO.	Parts Name
01	Hot water reservoir
02	Cold water reservoir
03	Fluid water pump
04	Cold water pump
05	Digital flow meter
06	Digital flow meter
07	Heating coil
08	Cooling spray pipe
09	Test section copper tube
10	Air duct
11	Axial Flow blower
12-13	Drain
14-20	Control valve

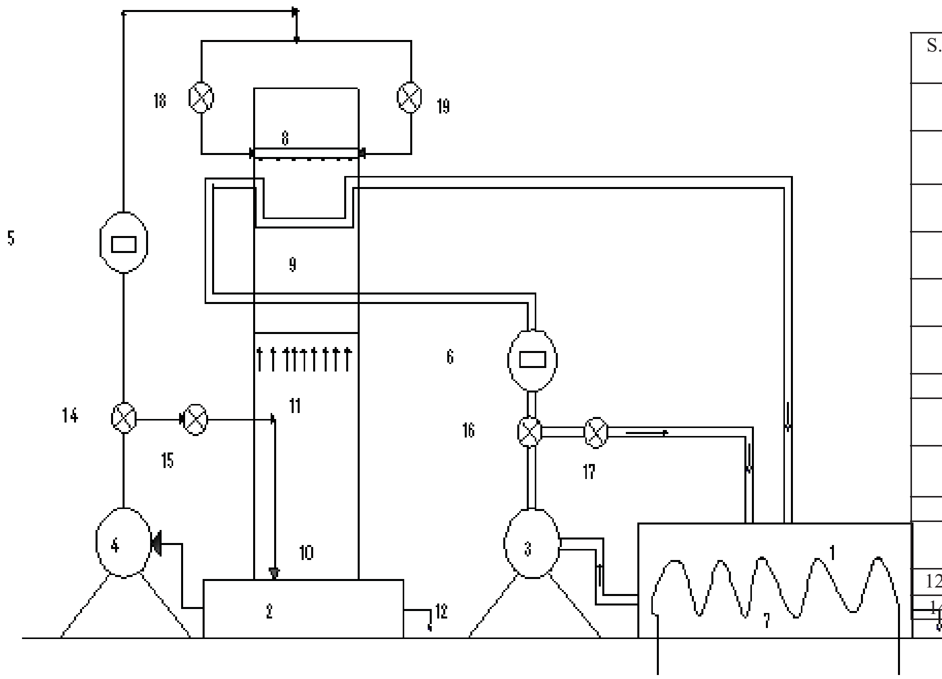

Figure (A) Line diagram of Experimental Set-up

IV. MATHEMATICAL FORMULATION & EQUATIONS

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

$$Q_w = m_p c_p (t_1 - t_2) \quad (2)$$

Reynold number of cooling water

$$R_{e,w} = 4\Gamma / \mu_w \quad (3)$$

Mass flow rate of cooling water:

$$\Gamma = m_w / 2l \quad (4)$$

The heat dissipation rate from the process fluid flowing inside the tube to air and water flowing over the tube:

$$Q_{wa} = m_p c_p (t_1 - t_2) \quad (5)$$

The Reynolds number of air is given by:

$$Re,a = \rho_a v_a D_o / \mu_a \quad (6)$$

The onset and permanent dryout mass transfer coefficient is:

$$K_{on} \text{ or } Kp = Q_{wa} / A_o (i_{s,tc,av} - i_{ai}) \quad (7)$$

where, $(i_{s,tc,av} - i_{ai})$ is the difference of enthalpies at saturated and unsaturated air.

The dimensionless enthalpy potential can be given as:

$$(EP)_{dl} = (i_{s,tc,av} - i_{ai}) / i_{fg} \tag{8}$$

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

According to the literature (Yung,1980), in the thick film mode, the diameter of a thick film mode d_t is related to liquid flow rate, physical properties off the fluid and tube spacing as:

$$d_t = 3^{1/4}(16\, \Gamma\, \sigma^{1/2} g^{-1} \rho^{-3/2} s^{-1/2})^{1/2} \tag{9}$$

And in the same way the droplet mode, the diameter of droplet is:

$$d_d = 3(\sigma\, \rho^{-1} g^{-1})^{1/2} \tag{10}$$

V. ONSET AND PERMANENT DRY PATCHES

In order to study the qualitative effects of Reynolds number of water ($R_{e,w}$) on the onset and permanent dryout heat fluxes (q_{on} & q_p), the computed results are shown in the graphical in Fig. 1 and 2 for a inlet temperature of process fluid (t_1).

Figures 1 and 2 depicts that both q_{on} and q_p increases with the $R_{e,w}$ and t_1. It can be concluded that at greater flow rates of cooling water, the tube surface tends to remain wet and there was formation of water film. However to breakdown this water film, the flow rate of process fluid is required to be increased. In this way, q_{on} or q_p gets increased with $R_{e,w}$. The Figures [3-6] shows the effects of Reynolds number of water on the onset and permanent dry out heat flux with some selected values of Reynolds number of air. The values of onset and permanent dry out heat flux, onset and permanent mass transfer coefficient increases with Reynolds number of water it is due to the fact that as Reynolds number of water increases there is more water evaporated from the surface and more heat flux required to form dry patches on to the surface of the tube. The quantitatively values of onset dry out heat fluxes for Reynolds number of air 1551±3.0, 3088±30, 4620±44, 6126±48 get enhanced by 35.96% to 514.04%, 61.63% to 542.48%, 57.80% to 531.40%, 12.4% to 312.12%, respectively and values of permanent dry out heat flux get enhanced by 27.72% to 218.87%, 32.88% to 374.38%, 37.10% to 307.10%, 50% to 416.31%. Similarly, the quantitative values of onset mass transfer coefficient for Reynolds number of air 1551±13, 3088±30, 4620±44, and 6126±48 get enhanced by 28.57% to 407.14%, 60% to 502.70%, 44.40% to 477.80%, and 15.80% to 278.95% respectively and values of permanent mass transfer coefficient for Reynolds number of air 1546±15, 3097±24, 4606±35, 6126±37 enhanced by 25.61% to 196.83%, 32.74% to 345.87%, 39.74% to 253.29%, 48.42% to 378.98%.

The results shows that for Re,w = 29.1±2, q_{on} get increased with Re,a by 16.8% to 173.97%, for Re,w= 149±3 get increased with Re,a by 38.80% to 126.20%, for Re,w = 326±6 get increased with Re,a 36.30% to 71.90%, for Re,w = 563±16 get increased with 22.72% to 92%, for Re,w = 855±44 get increased with Re,a 22% to 83.60%. Similarly, the variations of k_{on}, q_p, and k_p with Re,a are shown in Figs. [7-10].

The Figures [11-12] shows the variation q_p and k_p with $(EP)_{dl}$ with corresponding values of Re,a. As the flow cooling water increases the values of $(EP)_{dl}$ decreases and it would increased when cooling water flow rate decreases. From these it is concluded that cooling effect is increased when air velocity is increased.

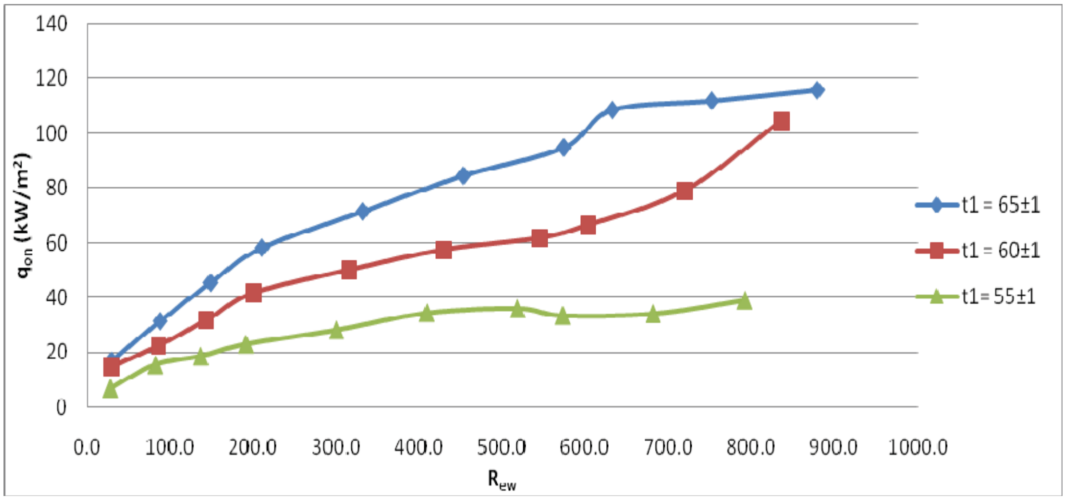

Figure1. Effect of film Reynolds number on the onset dryout heat flux of a single tube with only water flow

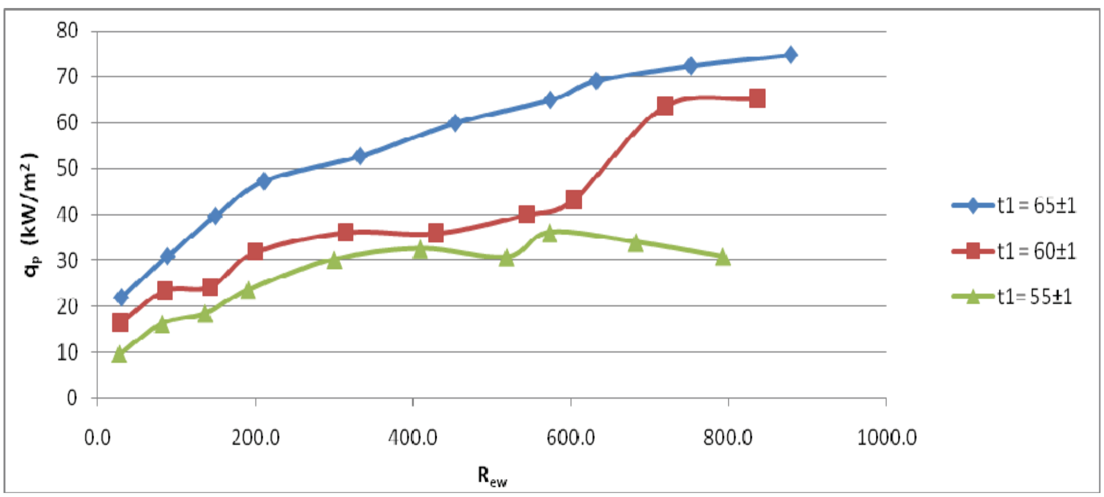

Figure2. Effect of film Reynolds number on the permanent dryout heat flux of a single tube with only water flow

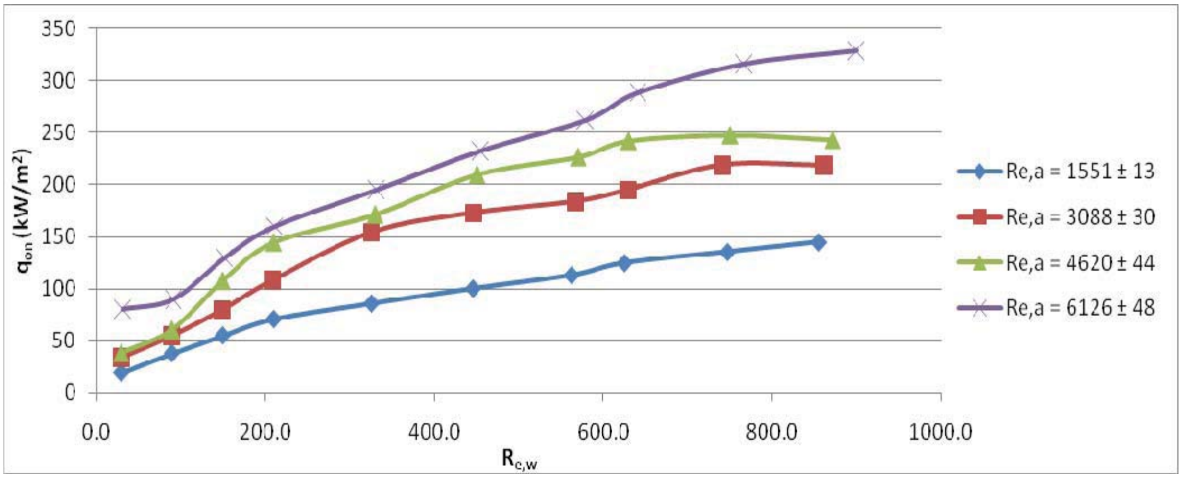

Figure 3. Effect of film Reynolds number on the onset dryout heat flux of a tube with simultaneous flows of water and air

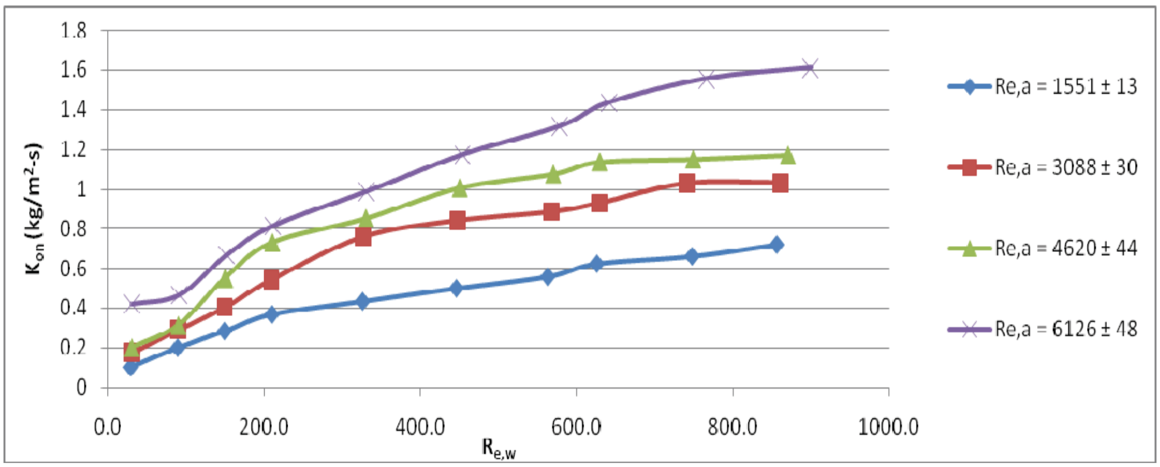

Figure 4. Effect of film Reynolds number on the onset dryout mass transfer coefficient of a tube with simultaneous flows of water and air

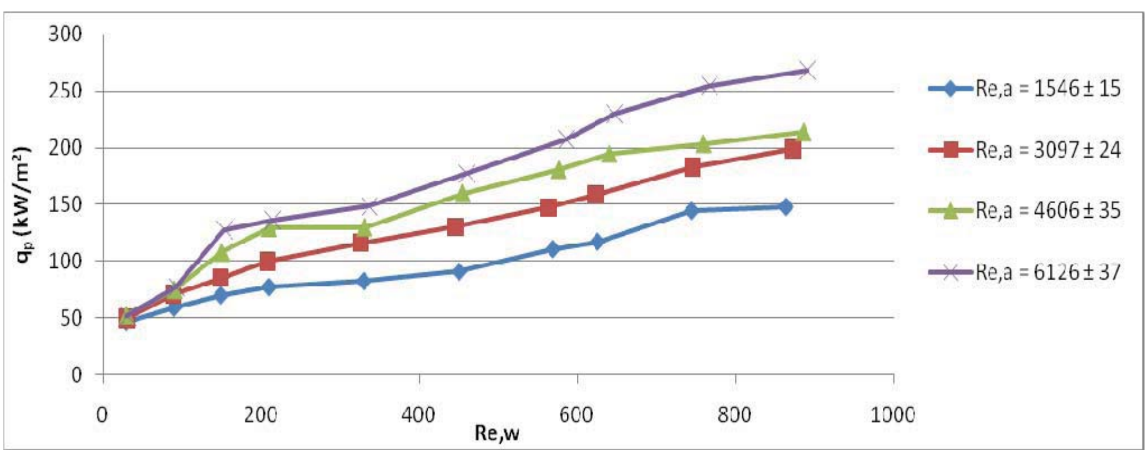

Figure5. Effect of film Reynolds number on the Permanent dryout heat flux of a tube with simultaneous flows of water and air

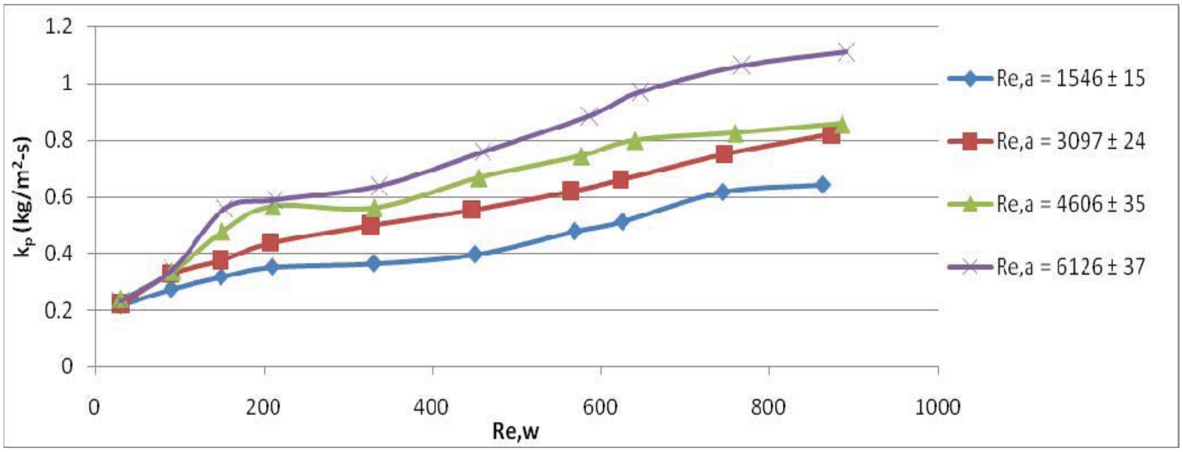

Figure6. Effect of film Reynolds number on the Permanent dryout mass transfer coefficient of a tube with simultaneous flows of water and air

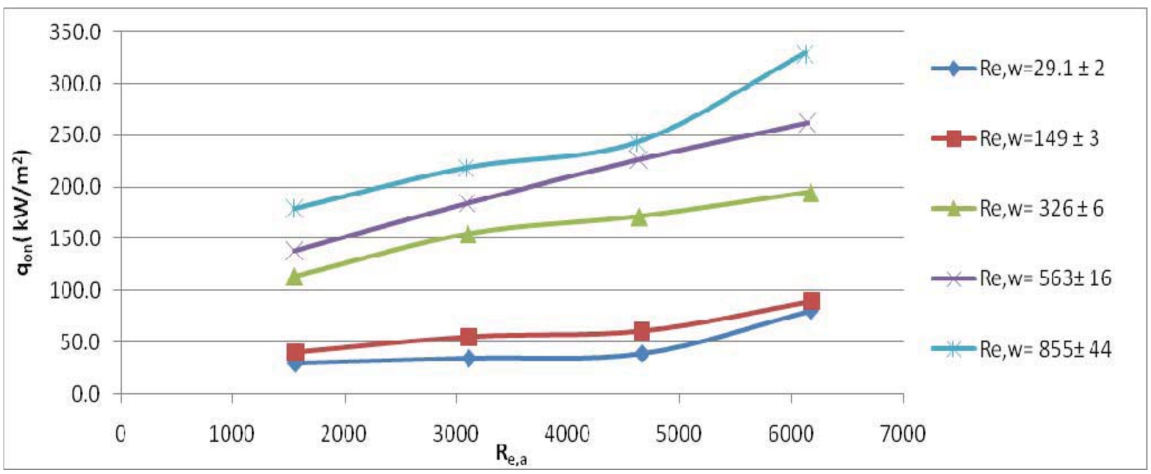

Figure7. Effect of air Reynolds number on the onset dryout heat flux of a tube with simultaneous flows of water and air

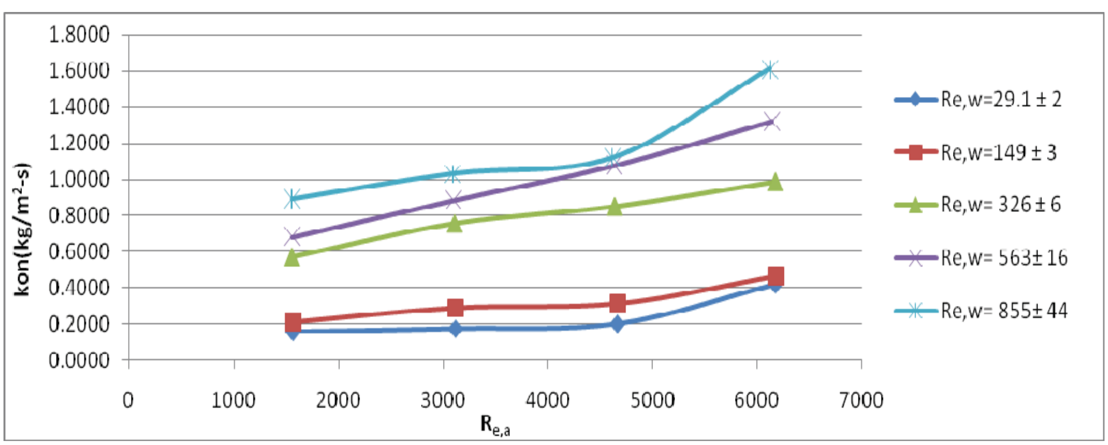

Figure8. Effect of air Reynolds number on the onset dryout mass transfer coefficient of a tube with simultaneous flows of water and air

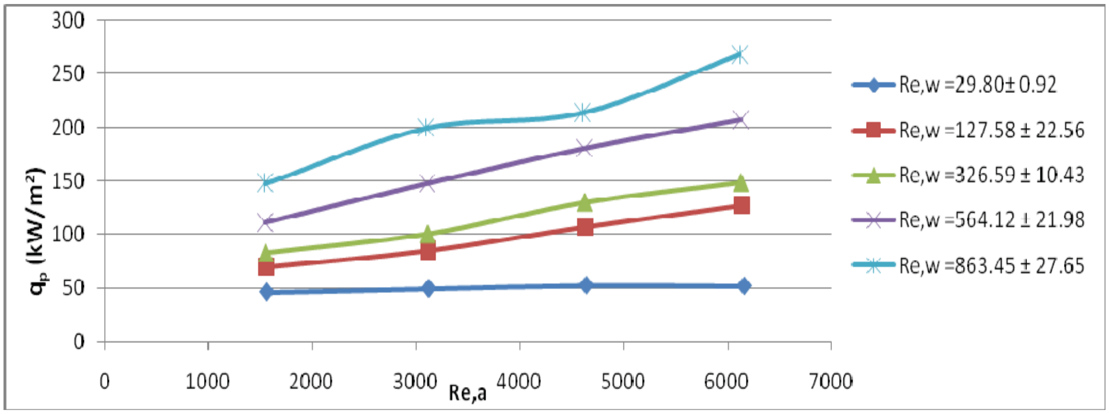

Figure9. Effect of air Reynolds number on the permanent dryout heat flux of a tube with simultaneous flows of water and air

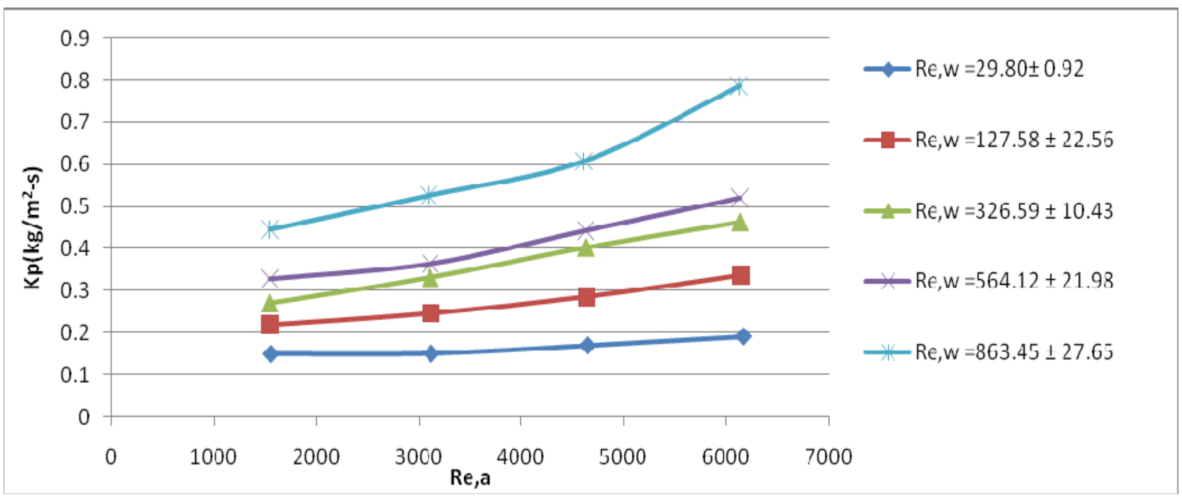

Figure10. Effect of air Reynolds number on the permanent dryout mass transfer coefficient of a tube with simultaneous flows of water and air

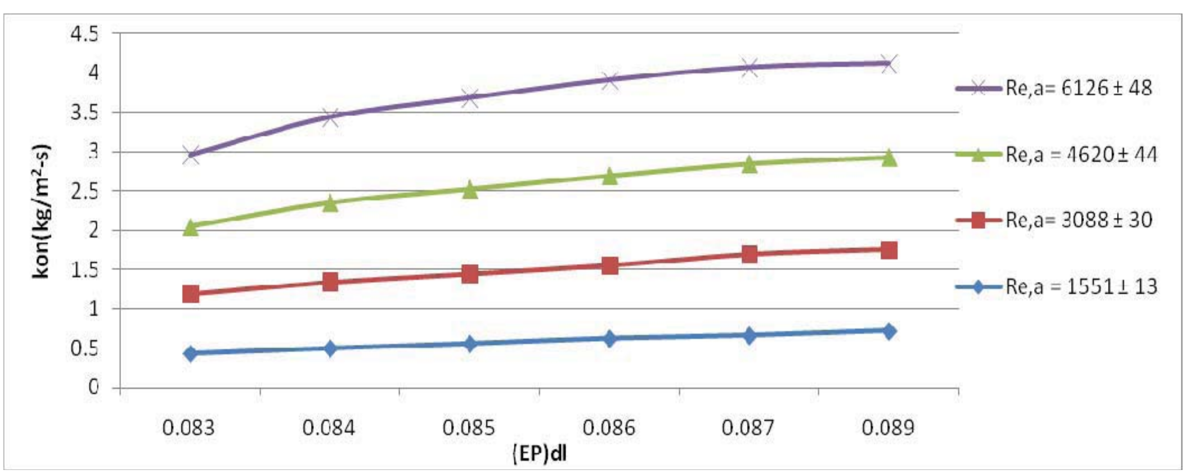

Figure11. Effect of dimensionless enthalpy potential on the onset dryout mass transfer coefficient of a tube with simultaneous flows of water and air

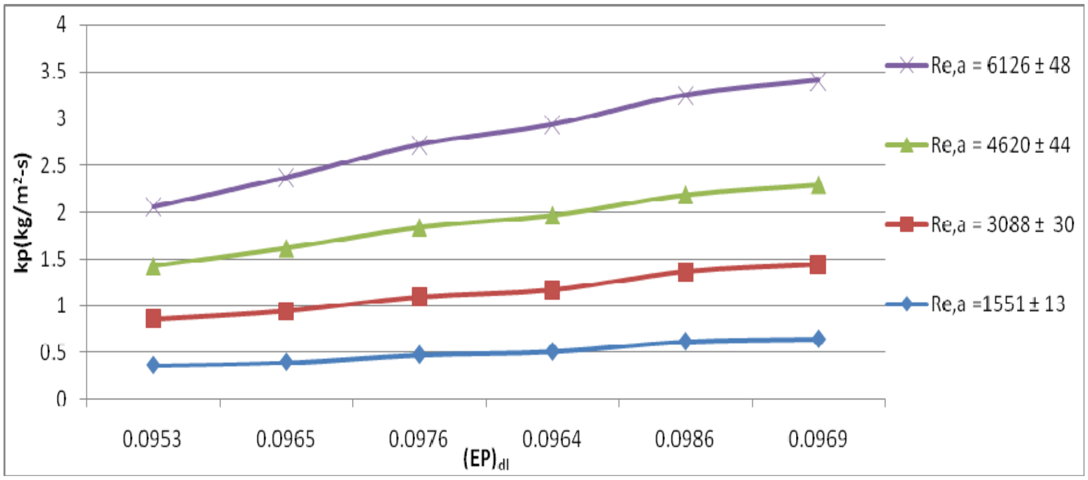

Figure12. Effect of dimensionless enthalpy potential on the permanent dryout mass transfer coefficient of a tube with simultaneous flows of water and air

V. CONCLUSIONS

The onset and permanent dry out heat flux increases with Reynolds number of water. It is because of the fact that higher the flow rate of cooling water, grater would be the heat flux required to evaporate water at faster from the surface of the tube to form dry patches on the surface of the tube. This is case of only water flow.

In case of both water and air flow it is observed that dry patches occur on the surface on the tube when Reynolds number of air increased. At higher flow of cooling water there is no dry patches formed on the surface of the tube.

Nomenclature

$A_\circ$ Outside area of the test tube $\left(m^2\right)$

c_p Specific heat of water at constant temperature, J/kgK

D Outer diameter of test tube (m)

d_j Diameter of jet

d_d Diameter of droplet

$(EP)_{dl}$ Dimensionless Enthalpy Potential

$i_{s,tc,av}$ Enthalpy of saturated air (J/kg)

i_{ai} Enthalpy of unsaturated air (J/kg)

i_{fg} Latent heat of vaporization at inlet temperature of cooling water (J/kg)

k_{on} Onset mass transfer coefficient $\left(kg/m^2-s\right)$

k_p Permanent mass transfer coefficient $\left(kg/m^2-s\right)$

l Effective length of test tube (m)

m_p Flow rate of process fluid (kg/s)

m_w Flow rate of cooling water (kg/s)

Q_{wa} Heat dissipation with water and air (W)

Q_w Heat dissipation with water (W)

Re_a Reynold number of air

Re_w Reynold number of water

s Tube spacing

Greek Symbols

Γ Mass flow rate per unit length, kg/s
σ Surface Tension
ρ Density
μ Dynamic viscosity

Subscripts

dl Dimensionless

p process fluid

w water

wa water and air

1 inlet

2 outlet

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