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Experimental Investigation on the Permanent Dryout Heat Flux and Dryout Mass Transfer Coefficient on Single Horizontal Brass Tube of an Evaporative Heat Exchanger

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Abstract*— An investigation on the permanent dryout heat flux and permanent dryout mass transfer coefficient on single horizontal brass tube of an evaporative heat exchanger is presented in this article. The permanent dry patch experiments were performed over the range of Reynolds number of water from 205 to 1015 when the tube is subjected to first type of flow condition i.e. water only; within the range of Reynolds number of water (Re_w) from 205 to 1010 and Reynolds number of air (Re_a) from 2475 to 8150 when the tube is subjected to second type of flow condition i.e. simultaneous flow of water and air. It is observed that the percent increase in dryout heat flux ratio with Reynolds number of water (Re_w) is found as 59.5 % when the tube is subjected to water only; The percent increase in dry out heat flux ratio with Reynolds number of water (Re_w) is found as 53.7 % when the tube is subjected to simultaneous flow of water and air. The permanent dryout mass transfer coefficient ratio first increases with Reynolds number of water (Re_w) and then decreases. Correlations are developed by using multiple regression analysis of experimental data. The correlations presented may prove to be useful in the improvement of design of an evaporative heat exchanger.

***Keywords*— permanent dryout heat flux, permanent dryout mass transfer coefficient, horizontal brass tube, heat exchanger, drypatch**

I. INTRODUCTION

Evaporative heat exchanger is widely used for cooling of heat process fluids in power generation plants, chemical plants, steel plants, petroleum refineries, nuclear power plants, air conditioning and refrigeration industries etc. In an evaporative heat exchanger, the cold water flows over horizontal tubes, inside which hot water passes. The hot water in the tubes undergoes evaporative cooling by the air blown from air duct from bottom to top. On the other hand falling film evaporators are based on heat transfer process that takes place when the cooling water is flowing downwards, due to gravity, on to a heated tube bundle. Falling film evaporators present several advantages in terms of higher cycle efficiency, reduced costs and low energy consumption. If the flow rate of the liquid film is reduced sufficiently or if the amount of heat added to the surface is relatively high, the film will thin, break down and dry patches will appear. Dry patches may be of two types i.e. onset and permanent. The onset dry patches are those that are occurred temporarily and permanent dry patches are those that are not rewetted again by putting some drops of cooling water manually. Liquid film breakdown over a horizontal cylinder was investigated by Ganic E.N.et al [1]. The effects of hydrophilic surface treatment on evaporative heat transfer at the outside wall of various kinds of copper tubes was studied by H.Y. Kim et al [2]. Simplified model for indirect cooling tower was given by Pascal Statbat et al [3]. The effects of liquid load , evaporation boiling point, temperature difference and tube diameter on the heat transfer coefficients outside the horizontal tube was studied by Li Xu et al [4].Various researchers investigated the heat transfer coefficient experimentally. Adriana Greco et al [5] presented the experimental heat transfer coefficients and pressure drop results obtained during the evaporation of pure R22 and the azeotropic mixture R507 (R125–R143a 50%/50% in weight). The evaporation heat transfer coefficient of the refrigerant R-

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134a in a vertical plate heat exchanger was investigated experimentally by Emila Zivcovic et al [6]. M. Fatouh et al [7] determined the evaporative local heat transfer coefficients of R22 and R410A in electrically heated a smooth horizontal copper tube with inner diameter of 9.525 mm and length of 1000 mm. Hani H. Sait et al [8] studied the effects of different falling film forms and hot water flow rate, on the heat transfer coefficient. M.M. Rehman et al [9] presented the evaluation of heat transfer coefficients for the flow of R22 through internally grooved copper tubes. The onset of local dryout of a saturated falling film with nucleate boiling on a vertical array of horizontal plain tube was investigated by Gherhardt Ribatski et al [10]. The heat transfer enhancement of falling film evaporation on a horizontal brass tube was studied Mostafa M. Awad et al [11]. The experimental evaluation of the heat transfer coefficient in a vertical tube rising film evaporator was reported by Syed Naveed Ul Hasan et al [12]. Correlations was introduced by S. Jani [13] for the Nusselt (heat transfer) and Sherwood (mass transfer) numbers for a falling film of Li/Br solution over a single horizontal heated tube used in generator of absorption chillers. Investigations on mass transfer coefficient and evaporative effectiveness of evaporative tubular heat exchanger was represented by Rajneesh and Rajkumar [14]. Dryout heat fluxes related to the onset and permanent dry patches was determined by Vikas Chander et al [15] without the air flow in case of single tube. Mass transfer coefficient is determined with water and air flow from the same tube.

II. TEST RIG.

Figure 1 shows the schematic diagram for experimental test rig. The test unit i.e. single horizontal brass tube is prepared with the following specifications: i) The internal diameter of the tube is 0.0234 m; ii) The external diameter of the tube is 0.0254 m; iii)The active length of the tube is 0.6 m; iv) The horizontal projection of the tube is 0.6 m; v) Outside surface area of the test unit is 0.047877 m^2.

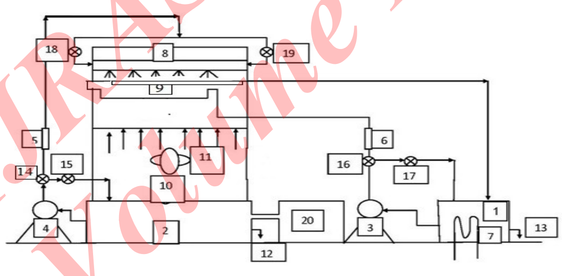

Fig.1 Sctematic diagram of an evaporative heat exchanger

1 Hot water tank; 2 Cold water tank ; 3 Hot water supply pump; 4 Cold water supply pump; 5-6 Digital flowmeter; 7 Insulated rods ; 8 Cooling water spray system ;9 Single horizontal brass tube;10 Air duct; 11 blower; 12-13 Drains; 14-19 flow control valves; 20 Feeder tank

In hot water circuit, five insulated rods (heating element) are used for heating of water in hot water tank. Then hot water is circulated into single horizontal brass tube through hot water supply pump. In cold water circuit, cold water from cold water tank is recirculated through cold water supply pump in cooling water spray pipe system. Then cold water is sprayed over single horizontal brass tube through which hot water is flowing. In an air circuit, air is entered at room temperature by axial flow blower. Then air from air duct flows from bottom to top of test section in which test unit is placed. The exit air moves out from top of the test section. RTD sensors are used to measure the temperature of hot water at inlet, outlet and at tube surface and then it is connected to the 32 channel programmable data logger, having resolution of 0.1^{o}C, that provides the temperatures at various points in the form of table directly. Calibrated digital flow meters, having resolution of 0.1 litre per minute, are used to measure the flow rates of cold water and hot water. Pre-calibrated digital anemometer, having resolution of 0.1 m/s, is used to measure velocity of exit air at top of test section. Flow control valves are used to control the flow rates of hot water and cold water throughout the test section.

III. METHODOLOGY

A. *Experimental Procedure for Permanent dryout heat flux (with water only)*

The cold water was made to flow over a single horizontal brass tube for about 60 hours until tube was fouled. The flow rate of cooling water varied from 5.07 x 10^{-2} kg/s to 24.48 x 10^{-2} kg/s with step increase of 4.06 x 10^{-2} kg/s. the initial value of flow rate was fixed slightly more than the minimum wetting rate (the minimum flow rate of cooling water required at which the unit length of brass tube becomes completely wet at no heat load). Then hot water was made to flow through the tube at 63 $\pm$ 0.25^{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. This condition of formation of dry patch is reffered to the condition of onset dry patch formation and corresponding heat load is termed as

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the onset dryout heat flux. Most of the dry patches were rewetted manually by putting some drops of cooling water by dropper over the surface of the tube. After getting disappearance of the dry patches from the 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 the tube. This condition of formation of dry patch is reffered to the condition of permanent dry patch formation and corresponding heat load is termed as the permanent dryout heat flux. At this state, the temperatures of hot water at the inlet, at the outlet and at the surface of tube were noted; also the temperatures of cooling water at the inlet of spray pipe was noted. The experiment was repeated for second, third, fourth, fifth and sixth reading of mass flow rate of cold water.

B. *Procedure for Permanent dryout heat flux and mass transfer coefficient (with simultaneous flow of water and air)*

The above experiment was repeated for first, second, third, fourth, fifth and sixth set of readings at air velocities at the top of test section as 1.5 m/s, 3.4 m/s, 4.2 m/s and 4.9 m/s. Each set of observations contains four readings. The temperature of air (dry bulb temperature and wet bulb temperature) at inlet and outlet of the test section were measured in addition to the all the parameters measured in previous experiment with water only.

IV. RANGE OF OPERATING VARIABLES

The flow rate of hot water of hot water in the experiment is taken in the range of 1.0574 kg/s to 1.2563 kg/s and the flow rate of cold water is taken in the range of 0.0507 kg/s to 0.2448 kg/s. The temperature of hot water in the experiment (with water only) is taken as 63±0.25°C, the temperature of cooling water is taken as 29.5 ±0.2°C and Reynolds number of water is taken as 205 to 1015. The temperature of hot water in the experiment (with simultaneous flow of water and air) is taken as 63±0.38°C, the temperature of cooling water is taken as 29.52 ±0.12°C, the temperature of the air is taken as 30 ±0.54°C, the velocity of the air is controlled in the range of 1.5 m/s to 4.9 m/s Reynolds number of water is taken in the range of 205 to 1010 and Reynolds number of air is taken in the range of 2475 to 8150.

V. MATHEMATICAL EQUATIONS

Heat dissipation rate (kW) and heat flux (kW/m^2) from the tube (when the tube is subjected to water only) can be written as:

$$Q_w = W_h C_{pw}(T_{hi} - T_{ho}) \qquad \text{(i)}$$

$$q_{(p)w} = \frac{Q_w}{A_o} \qquad \text{(ii)}$$

Heat dissipation rate (kW) and heat flux (kW/m^2) from the tube (when the tube is subjected to simultaneous flow of water and air) can be written as:

$$Q_{wa} = W_h C_{pw}(T_{hi} - T_{ho}) \qquad \text{(iii)}$$

$$q_{(p)wa} = \frac{Q_{wa}}{A_o} \qquad \text{(iv)}$$

where A_o is the outside surface area of the tube can be calculated as 0.047877 m^2.

Reynolds number of air and Reynolds number of water can be written as:

$$Re_a = \frac{\rho_a V_{t,s} D_o}{\mu_a} \qquad \text{(v)}$$

$$Re_w = \frac{4\Gamma}{\mu_w} \qquad \text{(vi)}$$

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

$$\Gamma = \frac{W_w}{2L} \qquad \text{(vii)}$$

where L is active length of tube and is taken as 0.6 m.

Permanent dryout heat flux ratio is the ratio of permanent dryout heat flux and maximum value of permanent dryout heat flux when the tube is subjected to water only can be written as:

$$\bar{q}_{(p)w} = \frac{q_{(p)w}}{q_{(p)w,max}} \qquad \text{(viii)}$$

Permanent dryout heat flux ratio is the ratio of permanent dryout heat flux and maximum value of permanent dryout heat flux when the tube is subjected to simultaneous flow of water and air can be written as:

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$$\bar{q}_{(p)wa} = \frac{q_{(p)wa}}{q_{(p)wa,max}} \quad \text{(ix)}$$

Mass transfer coefficient (kg/m^{2}s) pertaining to permanent condition of dry patches when the tube is subjected to simultaneous flow of water and air can be written as:

$$K_{(p)wa} = \frac{Q_{wa}}{A_o(i_s - i_a)} \quad \text{(x)}$$

where $(i_s - i_a)$ is enthalpy potential the difference of enthalpy of saturated air at the average of tube surface temperature and enthalpy of air at the inlet of heat exchanger.

Permanent dryout mass transfer coefficient ratio is the ratio of permanent dryout mass transfer coefficient and maximum value of permanent dryout mass transfer coefficient when the tube is subjected to simultaneous flow of water and air can be written as:

$$\bar{K}_{(p)wa} = \frac{K_{(p)wa}}{K_{(p)wa,max}} \quad \text{(xi)}$$

Dimensionless Enthalpy potential can be calculated as:

$$\overline{EP} = \frac{(i_s - i_a)}{i_{fg}} \quad \text{(xii)}$$

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

VI. RESULTS AND DISCUSSION

Fig. 2 shows that the effect of Reynolds number of water (Re_w) on the permanent dryout heat flux ratio ($\bar{q}_{(p)w}$) when the test unit is subjected to water only. The fig. shows that due to increase in the value of Reynolds number of water (Re_w), the dryout heat flux ratio ($\bar{q}_{(p)w}$) is increased upto the value of of Re_w=883 and then slightly drops. The effect of Reynolds number of water (Re_w) on the permanent dryout heat flux ratio ($\bar{q}_{(p)wa}$) for some selected value Reynolds number of air (Re_a) of when the test unit is subjected to simultaneous flow of water and air is shown in fig. 3. It is found from fig. that the dryout heat flux ratio ($\bar{q}_{(p)wa}$) increases with Reynolds number of water (Re_w). The dryout heat flux ratio ($\bar{q}_{(p)wa}$) increased upto the value of Re_w =877 and then slightly drops. It is because of the fact that higher the rate of cooling water being sprayed over the tubes, greater becomes the heat flux required to evaporate of water at faster rate fom the tube surface to give the appearance of dry patches. The effect of Reynolds number of water (Re_w) on the permanent dryout mass transfer coefficient ratio ($\bar{K}_{(p)wa}$) for some selected value Reynolds number of air (Re_a), when the test unit is subjected to simultaneous flow of water and air is shown in fig. 4. It is found that the dryout mass transfer coefficient ratio ($\bar{K}_{(p)wa}$) is increased upto the value of Re_w= 710 for Re,a=2489.92 $\pm$ 2.93 and 710.81 for Re,a =6973 $\pm$ 11.4 and then start decreasing. It is becuse of the fact that at lower value of Reynolds number of water, boundry layer of water film becomes thick. As we increases the Reynolds number of water it will shear off the boundry layer of water. The effect of dimensionless enthalpy potential ($\overline{EP}$) on permanent dryout mass transfer coefficient ratio ($\bar{K}_{(p)wa}$) for some selected value Reynolds number of air (Re_a) is shown in fig. 5, when the test unit is subjected to simultaneous flow of water and air. It is found that the dryout mass transfer coefficient ratio ($\bar{K}_{(p)wa}$) decreases with increase in dimensionless enthalpy potential ($\overline{EP}$). The dryout mass transfer coefficient ratio ($\bar{K}_{(p)wa}$) is decreased upto the value of $\overline{EP}$ = 0.10851 for Re,a = 5650$\pm$0.99.

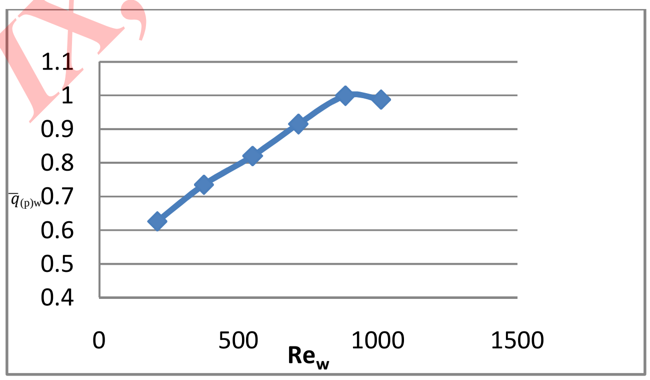

Fig.2 Effect of Reynolds number of water on the permanent dryout heat flux with water only

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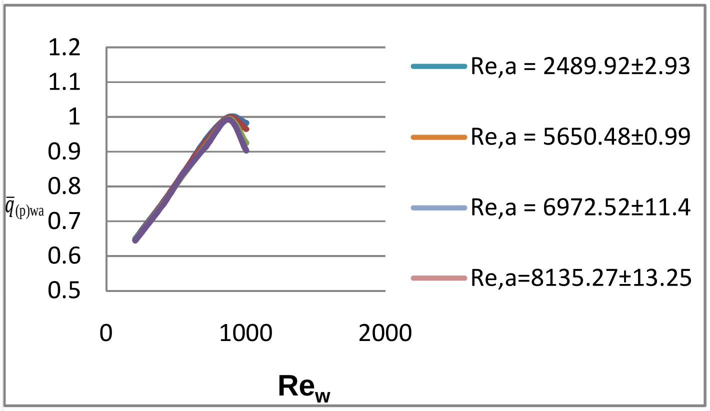
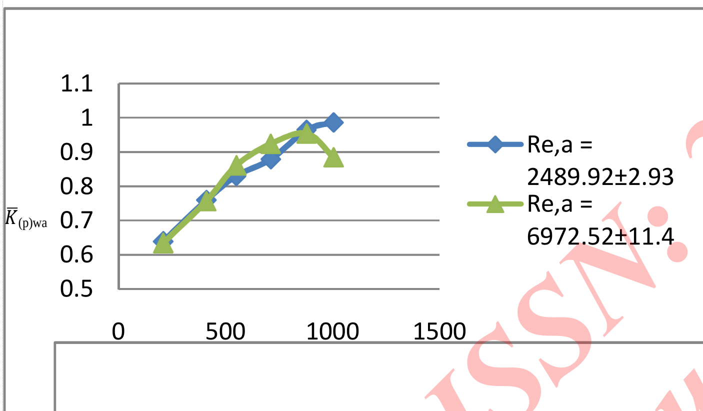


Fig. 3 Effect of Reynolds number of water on the permanent dryout heat flux with simultaneous flows of water and air



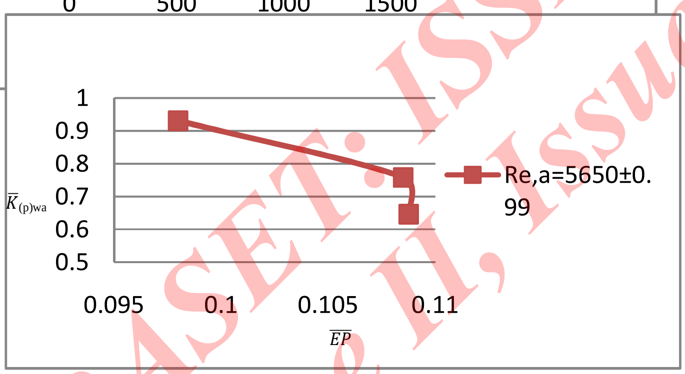


Fig. 4 Effect of Reynolds number of water on the permanent dry out mass transfer coefficient with simultaneous flows of water and air

Fig. 5 Effect of dimensionless Enthalpy Potential on the permanent dry out mass transfer coefficient with simultaneous flows of water and air

VII. MULTIPLE REGRESSION ANALYSIS: DEVELOPMENT OF CORRELATIONS

The permanent dryout heat flux with Reynolds number of water with the mean and standard deviation of -0.00021% and 0.018189% respectively are correlated as: For $205 \le Re_w \le 1015$, when hot water inlet temperature was $63 \pm 0.25^{o}C$

$$\bar{q}_{(p)w} = 0.12(Re_w)^{0.31} \quad \text{(xiii)}$$

This correlation shows a good agreement between experimental and predicted values of permanent dryout heat flux with an error of ±2.1%.

The permanent dryout heat flux with Reynolds number of water and Reynolds number of air with the mean and standard deviation of -0.00063% and 0.032107% respectively correlated as:

For $205 \le Re_w \le 1010$; $2475 \le Re_a \le 8150$ when hot water inlet temperature was $63 \pm 0.38^{o}C$.

$$\bar{q}_{(p)wa} = 0.152\,(Re_w)^{0.29}\,(Re_a)^{-0.016} \quad \text{(xiv)}$$

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

The permanent dryout mass transfer coefficient with Reynolds number of water, Reynolds number of air and dimensionless enthalpy potential with the mean and standard deviation of -0.0005% and 0.029181%, respectively are correlated as:

For $205 \le Re_w \le 1010$; $2475 \le Re_a \le 8150$; $0.0979 \le Re_a \le 0.1157$ when hot water inlet temperature was $63 \pm 0.38\ ^{o}C$

$$\bar{K}_{(p)wa} = 0.187\,(Re_w)^{0.27}\,(Re_a)^{-0.016}\,(\overline{EP})^{0.048} \quad \text{(xv)}$$

This correlation shows a good agreement between experimental and predicted values of permanent dryout mass transfer coefficient with an error of ±9%.

VIII. CONCLUSIONS

- From this experimental investigation on the single horizontal brass tube of an evaporative heat exchanger when the tube is subjected to first type of

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flow condition i.e. water only and when the the tube is subjected to second type of flow condition i.e. simultaneous flow of water and air following conclusions are drawn:

- the Reynolds number of water (Re_w) varied from 205 to 1015, the increase in dryout heat flux ratio ($\bar{q}_{(p)w}$) is found upto the Reynolds number of water (Re_w) = 882, when the tube is subjected to water only.
- When the Reynolds number of water (Re_w) varied from 205 to 1010 by keeping Reynolds number of air constant at Re_a= 2489, the increase in dryout heat flux ratio($\bar{q}_{(p)wa}$) is found upto the Reynolds number of water (Re_w) = 877, when the tube is subjected to simultaneous flow of water and air.
- The quantitative effects of Reynolds number of water (Re_w) on permanent dryout heat fluxes are more pronounced as compared to the Reynolds number of air (Re_a).
- When the Reynolds number of water (Re_w) varied from 205 to 1015, the percent increase in dryout heat flux ratio ($\bar{q}_{(p)w}$) with Reynolds number of water (Re_w) is found as 59.5%, when the tube is subjected to water only.
- When the Reynolds number of water (Re_w) varied from 205 to 1010 by keeping Reynolds number of air constant at Re_a= 2489, the percent increase in dryout heat flux ratio ($\bar{q}_{(p)wa}$) with Reynolds number of water (Re_w) is found as 53.7%, when the tube is subjected to simultaneous flow of water and air.
- The dryout mass transfer coefficient ratio ($\bar{K}_{(p)wa}$) first increase with Reynolds number of water (Re_w) and then decreases.
- The correlations presented may prove to be useful in the design of an evaporative heat exchanger, where minimum wetting rate must be known to determine the minimum liquid recycling ratio for an evaporative heat exchanger.

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