SIMULATION OF INDIRECT EVAPORATIVE COOLER HEAT EXCHANGER AT IRAQI CONDITIONS

Khalid A. Joudi[1], Khawla N. Hmood[2]
Department of Mechanical Engineering
College of Engineering, University of Baghdad, Baghdad, Iraq

ABSTRACT

The investigation included PHE performance variation with heat exchanger dimensions, plate spacing, inlet air velocities and inlet air temperatures. The momentum and energy equations were solved in 3-dimensions by using FLUENT 5.3 software and heat exchanger performance parameters were extracted from the post processing of the numerical data. In addition, a 2-dimensional fluid flow and heat transfer numerical analysis for the crossflow PHE was carried out. A numerical code based on the finite difference method and the SIMPLE algorithm was developed to solve the governing equations. The result of the numerical study for PHE performance shows that for both air streams the maximum thermal gradient occurs at 0.5 m/s inlet air velocity while the minimum occurs at 5 m/s velocity. Furthermore; the greater thermal gradient for the both air streams occurs at 3 mm plate spacing and decrease progressively to the lowest gradient at 10 mm spacing. Also, the results indicated that indirect evaporative cooling could be applied to obtain suitable outlet air temperatures for low and medium values of wet-bulb temperatures in arid climates.

الخلاصة

تعنى هذه الدراسة بمعرفة السلوك الحراري لمبادل حراري من النوع الصفائحي المستعمل للتبريد الذي يكون فيه جريان الهواء الحار والبارد بصورة متعامدة . حيث ان الهواء الخارجي يمر في الممرات الافقية والعمودية للمبادل الصفائحي ويتم نثر الماء على الهواء المار في الممرات العمودية بالتوازي مع تيار الهواء لكي تحصل عملية التبخر التي هي السبب الرئيسي في تبريد الهواء المار في هذه الممرات ومن ثم يتبرد به هواء الممرات الافقية .

تتضمن النمذجة العددية لهذا المبادل دراسة تغير اداء المبادل مع تغير ابعاده من 300x300 mm الى 1000x1000 mm مع تغير البعد بين الصفائح من 3 mm الى 15 mm وتغير سرعة دخول الهواء اليه من 0.3 m/s الى 5 m/s كما استخدمت ظروف التصميم الخارجية للعراق لتمثيل تغير درجات الحرارة ونمذجة الاداء .

اجريت نمذجة اولية ثلاثية الابعاد من خلال حل معادلات الزخم والطاقة باستخدام برنامج (FLUENT 5.3) حيث تحصلت العوامل المؤثرة على اداء المبادل من النتائج العددية .

من بعد ذلك اجري تحليل ثنائي الابعاد لمبادل التبريد التبخيري غير المباشر وذلك بأستخدام برنامج اعد لهذا الغرض كتب بلغة FORTRAN . واعتمدت خوارزمية سمبل (SAMPLE Algorithm) وطريقة الفروقات المحددة (Finite Difference) في حل المعادلات الحاكمة . وقد جرى اختبار العوامل المؤثرة على الاداء الحراري مثل درجة حرارة البصلة الجافة ودرجة حرارة البصلة الرطبة لتياري الهواء باسلوب منهجي مبرمج .

---

[1] Prof. of Mech. Eng., Corresponding author
[2] Graduate student

بينت النتائج المتحصلة من هذه الدراسة ان اداء المبادل الحراري يتحسن مع ازدياد ابعاد المبادل وصغر المسافة بين الصفائح وسرعة دخول الهواء الأوطأ .

كذلك ، فأن التبريد التبخيري غير المباشر يمكن ان يوفر درجة حرارة تجهيز مناسبة في العراق حيثما تكون درجة حرارة البصلة الرطبة واطئة او متوسطة القيمة .

Keywords: plate heat exchanger; wet crossflow; convective heat mass transfer; air coolers.

INTRODUCTION

Indirect evaporative cooling (IEC) involves sensibly cooling a process air stream and keeping its water content unchanged. Cross flow plate heat exchangers are widely used in IEC (Mazza 1984).
A cross-flow heat exchanger was used sheets only 0.18 mm thick, such that their thermal resistance was very low. Pescod 1974 proposed an idealized minimum supply temperature from an IEC unit as equal to the wet bulb temperature of room air.
Said and Khassaf 1984 evaluated the use of direct and indirect methods of evaporative cooling for residential comfort in Iraq using a heat exchanger. They concluded that the IEC is superior to direct evaporative cooling (DEC) in Mousl and Baghdad where it is usually dry; and that the use of IEC may be extended to the humid southern part of the country around Basrah where DEC would perform poorly. Yellott and Gamero 1984 described different types of indirect evaporative cooling units and the psychrometric analysis related to these various IEC system types. Kettleborough and Hsieh 1983 described a counter flow indirect evaporative cooler with configuration of upward flow of the primary air, downward flows of secondary air and water. Numerical analysis was utilized to study the thermal performance of the unit.

Awishalim and Al-Shawi 1986 and Enwia 1986 introduced a crossflow plate heat exchanger for use as an indirect evaporative cooler for Iraq. The heat exchanger was alternate passages of Aluminum plates. Both workers carried out tests on a number of heat exchanger geometries and plate spacing of 6 mm and 3.5 mm.

The performance of IEC is affected by changes in primary and secondary air velocities and mass flow rates, wet-bulb temperature, altitude, and other factors (Peterson 1993). Joudi and Mehdi 2000 carried out a theoretical study into the application of indirect-direct evaporative cooling systems in fulfillment of the variable cooling load of a typical Iraqi dwelling. This study was based on a mathematical model given by Pescod for a plate-type heat exchanger to simulate the indirect evaporative cooler.

Abdul Jabar 2000 conducted a study into the evaluation of indirect evaporative-desiccant cooling system performance. The study used a computer simulation for the evaluation of four systems. These systems employed a plate heat exchanger for the indirect evaporative cooler. The results showed that the best of the suggested systems was the indirect evaporative-desiccant cooling system.

Adhikari 2004 investigated alternatives to vapour compression technology by using polymer plates in crossflow heat exchanger as indirect evaporative cooling, which he claimed, provided a cost effective and efficient system for all climates.

The present study consists of two parts. The first part used the FLUENT software, to evaluate the performance of crossflow PHE upon the variation in the inlet velocities, and inlet temperatures for the two air streams. The temperature and velocity distributions for the two air streams inside the PHE were obtained and would be used as input data in a developed 2-dimensional analysis. The second part, involved a numerical scheme in 2-dimensions for the wet surface plate heat exchanger. The scheme was developed by a FORTRAN program based on the finite difference method. The calculation included distribution of temperature and velocity for the two air streams for various exchanger dimensions, inlet air velocities and varying inlet air conditions.

FLUENT SOFTWARE ANALYSIS

Physically, a PHE in Fig. 1 is a set of parallel metal plates, which separate hot and cold fluid streams from each other by using a crossflow mode. Heat transfer resistance through the plates is small compared to the air stream boundary layer resistance on each side of the plates. The basic system is a heat exchanger where both fluid flows remain in the gaseous phase and no condensation occurs over the exchanger surfaces.

The full numerical solution of heat exchangers is computationally prohibitive because the flow and temperature fields must be simultaneously determined in two fluids and making full simulation of heat exchangers is practically nonexistent (Patankar 1980). Heat transfer and fluid flow in enclosed channels is usually analyzed in 3-dimensions. Therefore, it was suggested to employ the FLUENT package to carry out the 3-dimensional analysis.

The steady state governing equations for continuity, momentum and energy are solved using a segregated solver, which means that temperature and flow fields are segregated from each other and were solved by using FLUENT 5.3 software. The initial guesses for velocity and temperature fields were set to constant values over the entire computational domain. To obtain the flow and heat transfer solutions, the solver undertakes iteration until the convergence criterion is satisfied, which employed scaled residuals of the modified variables in the governing equations as the measure. In addition, the averaged fluid temperature was examined implicitly for convergence (to less than a 0.01% variation between iterations).

The physical model of the present work was simulated as flow in 3D geometry using quadrilateral/hexahedral grids. The assumptions made were as follows:

- Incompressible laminar flow.
- Steady state conditions.
- Newtonian fluids.
- Forced convection heat transfer.
- Coupled conduction/convection heat transfer.

The calculation domain consisted of six rectangular channels stacked one over the other to form the crossflow plate heat exchanger. The limitation to six channels depended on the CPU of the computer and the number of nodes that could be solved at the same time. Various dimensions for a square plate heat exchanger were employed. These included 300x300, 500x500, 700x700, 800x800, and 1000x1000 mm. Also, the spacing between plates for each dimension was varied and included 3, 4, 5, 7, 8, 10, and 15 mm. The hot air was set at 320 K and the cold air at 302 K. Aluminum was selected for the plate heat exchanger material. Also, several inlet velocities were used for the hot and cold fluids. Namely, 0.3, 0.5, 0.7, 1, 1.5, 2, 2.5, 3, 4, and 5 m/s.

Fig. 2 is a typical case for demonstrating the effect of plate dimensions on the hot air temperature distribution. It is clear that the thermal gradient increased progressively with increasing heat exchanger dimensions. Larger area means higher heat exchanger effectiveness. The effect of various plate spacing on the hot air temperature distribution is shown in Fig. 3. It is observed that the thermal gradient increased with decreasing plate spacing. This is due to the two reasons. First, Re number is decreasing. Second, heat transfer coefficient is increasing. Fig. 4 illustrates the effect of the various inlet air velocities on the hot air temperature distribution. It clearly shows that the thermal gradient increases with decreasing inlet velocity for the hot air. As a result the thermal gradient increased progressively with decreasing plate spacing, lower velocity and larger dimensions. As a result of, it is increasing in heat transfer between two streams.

It is intended to find a suitable IEC heat exchanger for domestic use at Iraqi houses. The most commonly used evaporative coolers are 2500, 3500, 4500 cfm (1179.875, 1651.825, 2123.775 l/s) air coolers. Dimensions of a suitable IEC heat exchanger and other conditions for such flow rates are sought after. The Fluent results would be used for this purpose in a 2-D analysis. It is observed from the results in Fig. 5, that the values of ΔT_h and ΔT_c for 700x700, 800x800, and 1000x1000 mm are comparable. The 800x800 mm is selected as representative of the three. Thus, the 300x300, 500x500 and 800x800 mm dimensions will be used for the purposes of this study in the 2-D analysis.

From Figs. 6, and 7, the 3 mm plate spacing gives the maximum values as a consequence of ΔT_h and ΔT_c. The 7 and 8 mm spacing give approximately the same results for ΔT_h and ΔT_c. Hence, 8 mm plate spacing may be selected as representative of both. The maximum value of plate spacing in heat exchangers used in air conditioning does not usually exceed 12.5 mm (ASHRAE 2000 equipment and applications). Therefore, the 15 mm plate spacing will be neglected in the 2-dimensional analysis. Thus, the representative plate spacings that will be used in the 2-dimensional analysis will be 3, 5, 8, and 10 mm. Also, from these Figures, it can be observed that velocities of 1.5 and 2 m/s give approximately the same value of ΔT_h and ΔT_c and the velocities of 2.5 and 3 m/s give the same results. Thus, the representative velocities that will be used in the 2-dimensional analysis will be 0.5, 1, 2, 3, 4, and 5 m/s.

TWO DIMENSIONAL SIMULATION MODEL FOR THE HEAT EXCHANGER

To simulate a plate heat exchanger, the velocity field is uncoupled from the temperature field and determined first before solving the energy equation. The continuity and momentum equations need to be solved to determine the velocity field. Since the momentum equations are nonlinear and coupled to the continuity equation, they have to be solved iteratively (Albakhit and Fakheri 2005).

The general assumptions for the problem solution used by Mishra, et al. 2004 which will be adopted here are as follows:-

1. Incompressible fluids and hydrodynamically developing flows.
2. Both fluids are single phase, unmixed and do not contain any volumetric source of heat generation.
3. The exchanger shell is adiabatic and the effects of the asymmetry in the top and bottom layers are neglected. Therefore, the heat exchanger may be assumed to comprise of a number of symmetric sections.
4. Variation of temperature in the fluid streams in a direction normal to the separating plate (z-direction) is neglected.
5. Conduction along the walls is negligible.

FLOW GOVERNING EQUATIONS IN TWO DIMENSIONS

The incompressible Navier-Stokes equations in 2-dimensions are in the following form:

- Continuity Equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \qquad (1)$$

- Momentum Equations:

$$\frac{\partial(\rho u^2)}{\partial x} + \frac{\partial(\rho uv)}{\partial y} = -\frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \qquad (2)$$

$$\frac{\partial(\rho vu)}{\partial x} + \frac{\partial(\rho v^2)}{\partial y} = -\frac{\partial p}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) \qquad (3)$$

One type of numerical solution for the above equations is an iterative process called the pressure correction technique. This technique is embodied in an algorithm called SIMPLE (semi-implicit method for pressure-linked equations) (Patankar 1980). The above equations are applied for the two fluids to find the velocity distribution along the flow direction, by non-dimensionalizing as follows (Albakhit and Fakheri 2005), (Incropera and DeWitt 1996);

- For the hot fluid,

$$\left.\begin{array}{lll} U = \dfrac{u}{u_{in}}, & V = \dfrac{v}{u_{in}}, & X = \dfrac{x}{L}, \\ Y = \dfrac{y}{L}, & P = \dfrac{P - P_{in}}{\rho u_{in}^2} & \end{array}\right\} \qquad (4)$$

These terms are substituted in eqs. (2) and (3). The resulting equations take the following form;

$$\frac{\partial U^2}{\partial X} + \frac{\partial UV}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right) \qquad (5)$$

$$\frac{\partial VU}{\partial X} + \frac{\partial V^2}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right) \qquad (6)$$

- For the cold fluid,

$$\left.\begin{array}{lll} U = \dfrac{u}{v_{in}}, & V = \dfrac{v}{v_{in}}, & X = \dfrac{x}{L}, \\ Y = \dfrac{y}{L}, & P = \dfrac{P - P_{in}}{\rho v_{in}^2} & \end{array}\right\} \qquad (7)$$

These equations will be used in the iterative solution.

TWO DIMENSIONAL WET HEAT EXCHANGER ANALYSIS

Evaporatively cooled heat exchangers (Fig. 8) can achieve heat transfer rates higher than dry heat exchangers. Heat transfer takes place from hot dry air, flowing inside channels, to wet air in alternate channels, through a water film which is formed by spraying water onto the heat exchanger surface. Spray water is circulated in a closed circuit. Heat transfer to air is in sensible and latent forms. The latent heat transfer makes up a major part and is produced by the evaporation of a small amount of the spray water into the air stream. When compared with dry heat exchangers, wet heat exchangers can achieve lower temperatures because the air wet-bulb temperature is, theoretically, the ultimate limit of the air-water direct contact process.

Essentially, the water on the plate is "excessive water" in the wet passage. Because the thickness of the water film is small, it is assumed that the temperature difference across the water film on the plate is negligible, and, hence, the local temperature of water on the plate can be taken to be the same as the local temperature of the plate (Kettleborough and Hsieh 1983). For a steady state condition, it is assumed that spray water flow rate is sufficient to wet the whole surface of the plate. The analysis of the energy balance is as follows:-

Heat is transferred from the hot air to the spray water film through the plate thickness as a result of the temperature gradient between the hot air temperature T_h and the spray water temperature T_s. The rate of heat lost by the hot air dq_h is

$$dq_h = \dot{m}_h c_{ph}\, dT_h = -U_o\,(T_h - T_s)\, dA \qquad (8)$$

Where, U_o is the overall heat transfer coefficient between the hot air through the separating plate to the water film on the other surface. It is given as,

$$U_o = \frac{1}{1/h_h + s/k} \qquad (9)$$

To find the heat transfer coefficients for the two air streams the Nusselt number must be specified. For laminar, developed conditions with constant wall temperature and Reynolds number less than 2300, the Nusselt number is given as (Incropera and DeWitt 1996);

$$Nu = 7.542 + 7.542 * \left(0.003 + 0.039 * \frac{a}{b}\right) * \frac{Pe}{L/d_h} \qquad (10)$$

Where, Pe is the Peclet number, and for turbulent flow, where the Reynolds number is higher than 2300, it is given as,

$$Nu = 0.036\ \mathrm{Re}^{0.8}\ \mathrm{Pr}^{\frac{1}{3}} \left(\frac{L}{d_h}\right)^{-0.054} \qquad (11)$$

After the calculation of the Nusselt number, the value for local convection coefficient h can be determined from,

$$h = \frac{Nu\ k}{d_h} \qquad (12)$$

Heat transfer from the cold air–water interface region to the cold air stream dq_c consists of a sensible part dq_s and a latent part dq_l;

$$dq_c = \dot{m}_c\, dH_c = dq_s + dq_l \qquad (13)$$

Substituting for the sensible and latent heats gives,

$$dq_c = \dot{m}_c dH_c = h_i (T_i - T_c) + k_m (w_i - w_c) h_{fg} dA \qquad (14)$$

The enthalpy of air and water vapor mixtures is given as (Joudi 1990);

$$H = (1.005\,T_d - 0.026) + w(2501 + 1.84\,T_d) \qquad (15)$$

Then, the temperature of moist air can be written as

$$T_c = \frac{H_c + 0.026 - 2501\,w}{1.005 + 1.84\,w} \qquad (16)$$

The specific heat for humid air c_{pu} is equal to $(1.005 + 1.84w)$. Substituting for T_c and T_i from Eq. (16) and taken into consideration that for air and water mixtures the Lewis number could be taken as unity (ASHRAE fundamentals 1997) the Eq. (14) is reduced to :

$$dq_c = \dot{m}_c dH_c = k_m\,(H_i - H_c)\, dA \qquad (17)$$

The liquid side of the interface offers a negligible resistance to heat transfer, so that the interface enthalpy H_i in Eq. (17) could be considered equal to the saturated air enthalpy H_s at the spray water temperature T_s. Therefore, Eq. (17) is rewritten as;

$$dq_c = \dot{m}_c dH_c = k_m (H_s - H_c) dA \qquad (18)$$

Eq. (18) is called the Merkel equation (Hasan and Siren 2002). It shows that the energy transfer could be represented by an overall process based on enthalpy potential difference, between air–water interface and bulk air, as the driving force.
But, for the DEC the enthalpy is essentially constant. Therefore, the sensible heat transfer from cold air to the air-water interface, can be written as,

$$-\dot{m}_c\, c_{pu}\, dT_c = h_i\,(T_c - T_s)\, dA \qquad (19)$$

Also, $h_i = c_{pu}\, k_m$ then;

$$-\dot{m}_c c_{pu}\, dT_c = c_{pu}\, k_m\,(T_c - T_s)\, dA \qquad (20)$$

$$\dot{m}_c dT_c = k_m\,(T_s - T_c)\, dA \qquad (21)$$

In problems where both convection and mass transfer are important, the dimensionless ratio υ/D_{ab} is important and is called the Schmidt number Sc. Thus, the Schmidt number plays a role similar to that of the Prandtl number in convection heat transfer problems. The functional dependence of the heat transfer coefficient is;

$$\frac{h\,d_h}{k} = f(\text{Re}, \text{Pr}) \qquad (22)$$

In convection mass transfer problems, the functional dependence of the mass transfer coefficient is;

$$\frac{k_m\,d_h}{D_{ab}} = f(\text{Re}, Sc) \qquad (23)$$

The similarities between the governing equations for heat, mass, and momentum transfer suggest that empirical correlations for mass transfer coefficient would be similar to those for the heat transfer coefficient. The grouping of terms $k_m d_h / D_{ab}$ is called the Sherwood number Sh.

The mass transfer coefficient for the humid air could be found from the Sherwood number for laminar flow at constant wall temperature by the following Equation (Incropera and DeWitt 1996);

$$Sh = 7.542 + 7.542 * \left(0.003 + 0.039 * \frac{a}{b}\right) * \frac{\text{Re} * Sc}{L/d_h} \qquad (24)$$

For turbulent flow the Sherwood number is (Incropera and DeWitt 1996);

$$Sh = 0.036\ \text{Re}^{0.8}\ Sc^{\frac{1}{3}} \left(\frac{L}{d_h}\right)^{-0.054} \qquad (25)$$

$$k_m = \frac{Sh * D_{ab}}{d_h} \qquad (26)$$

Where, the value of the diffusion coefficient D_{ab} at one atmosphere is equal to $0.26 * 10^{-4} m^2/s$ (Table A-8, Incropera and DeWitt 1996). Also, the value of Sc, for water vapor and air, is equal to 0.6 (Table A-18, Kays and Crawford 1993).

The solution to the above equations is in two parts. The first part solves the flow equations (continuity and momentum) and the second part solves the energy equation.

COMPUTATIONAL MODEL FOR THE WET CROSSFLOW HEAT EXCHANGER (IEC)

The surface area of the heat exchanger is divided into a number of two-dimensional small regions. Thus, the flow in the X-direction is divided into (Nx) sections and flow in the Y-direction into (Ny) sections. The reference points form a nodal network, in which the value of a new point is calculated from the results of its adjacent points where, the properties of hot air, spray water and cold air are defined on the surface boundaries. The numerical solution was based on a finite difference representation for the eqs. (8) and (21).

The variation of spray water temperature along a channel is small. Therefore, it could be assumed that the spray water temperature is constant and equal to T_s. For the above equations the forward finite difference approach will be applied and the average at $(i+\frac{1}{2}, j)$ and $(i, j+\frac{1}{2})$ will be in the following form;

By using the finite difference method and the average of T_s, heat transfer from the hot air to the spray water in eq. (8) becomes;

$$\frac{1}{2} u_h \rho c_{ph} a_h\, dx \left(T_h(i,j) - T_h(i-1,j)\right) = U_o\, dx\, dy \left(\frac{T_s + T_s}{2} - \frac{T_h(i,j) + T_h(i-1,j)}{2}\right) \qquad (27)$$

After some rearrangement the above equation. can be written as;

$$T_h(i,j) = \frac{2 U_o\, dy}{u_h \rho c_{ph} a_h + U_o\, dy} T_s + \frac{u_h \rho c_{ph} a_h - U_o\, dy}{u_h \rho c_{ph} a_h + U_o\, dy} T_h(i-1,j) \qquad (28)$$

The heat transfer from air- water interface to the cold air in eq. (26) becomes;

$$\frac{1}{2} v_c \rho a_c\, dy \left(T_c(i,j+1) - T_c(i,j)\right) = k_m\, dx\, dy \left(\frac{T_s + T_s}{2} - \frac{T_c(i,j+1) + T_c(i,j)}{2}\right) \qquad (29)$$

After some rearrangement the above equation. can be written as;

$$T_c(i,j) = \frac{v_c \rho a_c - k_m\, dx}{v_c \rho a_c + k_m\, dx} T_c(i,j-1) + \frac{2 k_m\, dx}{v_c \rho a_c + k_m\, dx} T_s \qquad (30)$$

The FORTRAN code for solving the energy equation was composed of two parts. The first part reads the velocity field which was found from the solution of the flow equations for the hot and cold fluids. The second part of the code solved the energy equation for the entire heat exchanger and determined the temperature distribution. Also, the wet bulb temperature T_w was obtained from dry bulb temperature T_d and moisture content w for the outlet air from the heat exchanger by using the empirical relation given by Said and Khassaf 1984 as;

$$T_w = 2.265\left(1.97 + 4.3T_d + 10^4 w\right)^{\frac{1}{2}} - 14.85 \qquad (31)$$

RESULTS AND DISCUSSION

The result for the present work for the following conditions, the inlet temperature for hot air equal to 45 ^oC and the cold air temperature equal to 29 ^oC while the spray water temperature was set equal to the wet bulb temperature at a value of 22 ^oC. Figs. 9, 10, and 11 are taken as typical of the results. Fig. 9 shows that the thermal gradient increases with decreasing inlet velocity for the process air. It is important to note that, the inlet velocity affects the Re number, Nu number, the heat transfer coefficient and consequently the rate of heat transfer. However, the maximum thermal gradient occurs at 0.5 m/s inlet air velocity and the minimum thermal gradient occurs at 5 m/s velocity. As a result of, the lower velocity improves the heat transfer between the two air streams. This result is similar to that for the wet air passage but with opposite direction as shown in Fig. 10. Fig. 11 is typical for other results and shows that the thermal gradient increases with decreasing plate spacing. The definition of Re number in this work is based on the plate spacing ($d_h = 2a$). Nu number is a function of the Re number and renders a heat transfer coefficient, for the smaller spacing, higher in value than that for the larger plate spacing. Analysis of this result shows that the heat capacity rate is decreased using narrower passages because a lower air flow rate is obtained at any given velocity. Thereby, the greater thermal gradient occurs at 3 mm plate spacing and decreases progressively to the lowest gradient at 10 mm spacing. Also, these Figures show a uniform outflow temperature for the two air streams from the outflow port. This is because of the effect of assumed constant plate surface temperature. The quantative effects of inlet air velocity and plate spacing on the wet heat exchanger performance are further shown in Figs. 12 to 15. These figures represent the temperature difference between the inlet and average outlet process air temperatures, the difference between the inlet and average outlet wet passage air temperatures and the difference between the inlet and average outlet wet bulb temperature for the process air with the inlet velocity of the two air streams. These figures are typical of the results for other plate heat exchanger dimensions. It is clear that ΔT_d, for the two air streams, and ΔT_w, for the process air, for any plate spacing, decrease with increasing inlet air velocity. The maximum value of temperature difference for each stream occurs for 3 mm plate spacing and at 0.5 m/s inlet velocity. The minimum value of temperature difference occurs for 10 mm plate spacing and at 5 m/s inlet velocity. This may be due to the larger number of plates used for the 3 mm spacing case where, the number of plates is obtained by dividing the dimension of plate on the spacing between plates. Also, ΔT_d for the two air streams is equal for any one specific condition because of the thermal balance between the two air streams. Moreover, ΔT for both streams becomes more pronounced as the plate heat exchanger dimensions increases from 300x300 to 800x800 mm as shown in Figs. 12 to 15.

Figs. 12 and 13 show that for 5 mm spacing, ΔT at 5 m/s velocity is larger its value at 4 m/s velocity and the curve changes direction. This may be due to turbulence setting in at 5 m/s velocity, which raises the rate of heat transfer between the two air streams. For 8 and 10 mm plate spacing, the slope of the curves is steep at low velocity and becomes nearly horizontal and steady after a velocity of 3 m/s. Figs. 14 and 15 show similar results for 800x800 mm heat exchanger dimensions. Change in direction of the curves is more pronounced in this case.

The previous results were for a specified dry-bulb and wet-bulb temperature test program. They were carried out at restricted Re numbers for each spacing by restricting the inlet air velocity. If Re number is less than 2300 the flow is laminar, if greater or equal to 2300 the flow is turbulent (Incropera and DeWitt 1996). Further tests were carried out at velocities ranging from 0.5 to 9 m/s to find out when turbulence sets in. The values of such velocities, as obtained for 800x800 mm PHE dimension from Fig. 16, for each plate spacing are listed in Table (1);

Table 1. Inlet air velocity at which turbulence occurs

Spacing mm	Inlet air velocity m/s

3	7
5	5
8	3
10	3

The effect of heat exchanger dimensions on the process air and the wet air passage temperature is shown in Fig. 17 for an inlet air velocity of 2 m/s and 5 mm plate spacing. The maximum thermal gradient occurs in the 800x800 mm heat exchanger because of the increased heat transfer area. Fig. 19 shows a comparison between experimental results for a 200x200 mm IEC heat exchanged obtained by Enwia 1986 and the results of the present model. Dimensions and parameter values employed by Joudi and Mehdi 2000 Ref. were used in the present model. Namely 3.5 mm plate spacing and an inlet velocity for the two streams varying from 1.1 to 6.7 m/s. Good agreement is observed in Fig. 18A. This comparison is repeated for 6 mm plate spacing and an inlet air velocity range from 0.77 to 7.6 m/s in Fig. 18B. Fig. 19 compares the present results with that of a parallel flow plate heat exchanger by Kettleborough and Hsieh 1983. The crossflow is difference from the parallel flow. Where, the temperature in outlet crossflow is variable along the outlet while, in the parallel flow the outlet temperature is equality.

CONCLUSIONS

1. The process air outlet temperature from PHE is lower with increased heat exchanger dimensions, lower inlet air velocity and smaller plate spacing.
2. The present finite difference model is in good agreement with other experimental works and the errors in the results not exceeded 15%.
3. The indirect evaporative cooler PHE rendered substantial reductions in dry bulb temperature and can be used effectively in hot arid climates as in Iraq.

REFRENCES

- ASHREA Handbook of Fundamentals, 1997.
- ASHRAE 2000 equipment and applications.
- Awishalim, J., and Al-Shawi, I.M., 1986, "Development and assessment of a plate heat exchanger for indirect evaporative", 4th conference of the Scientific Research Council, Iraq.
- Abdul Jabar, M., 2000, "Simulation of Indirect Evaporative Desiccant Cooling System", M.Sc. Thesis, Mechanical Engineering Department, University of Baghdad.
- Adhikari, A.K., 2004, "Improving energy efficiency using indirect evaporative cooling with vapour compression assistance", M.Sc. Thesis, The Australian Institute of Energy, Sydney Branch. 2004.
- Albakhit, H., and Fakheri, A., 2005, "A Hybrid Approach for Full Numerical Simulation of Heat Exchangers", ASME Heat Transfer Summer Conference, July 17-22, 2005, San Francisco, CA, USA.
- Enwia, J.A., 1986, "Development and Assessment of a Crossflow Plate Heat Exchanger for Use in Indirect Evaporative Cooling", M. Sc. Thesis, Mechanical Engineering Department, University of Baghdad.
- Hasan, A., and Siren, K., 2002, "Theoretical and Computational Analysis of Closed Wet Cooling Towers and Its Applications in Cooling of Buildings", Energy and Buildings, Vol. 34, No. 5, pp. 477-486.
- Incropera, P.F., and DeWitt, D.P., 1996, "Fundamentals of heat and mass transfer", 4th Ed.: John Wiley & Sons, Inc. Pp. 886. ISBN 0-471-30460-3
- Joudi, K.A., 1990, "Principles of Air-conditioning and Refrigeration", Basrah, University Press, 2nd printing, 1990, In Arabic.
- Joudi, K.A., and Mehdi, S.M., 2000, "Application of Indirect Evaporative Cooling to a Variable Domestic Cooling Load", Energy conversion and Management, Volume 41, Issue 17, 1 November 2000, Pages 1931-1951.
- Kays, W.M., and Crawford, M.E., 1993, "Convective Heat and Mass Transfer", Third Edition, McGraw-Hill Book Co.-Singapore.
- Kettleborough, C.F., and Hsieh, C.S., 1983, "The Thermal Performance of the Wet Surface Plastic Plate Heat Exchanger Used As an Indirect Evaporative Cooler", Trans. Of the ASME, J. of Heat Transfer, Vol. 105, pp. 366-372, May 1983.
- Mazza, V. 1984, "Plate heat exchangers for HVAC applications", Heating /Piping /Air Conditioning, Vol. 56, No. 8, pp. 59-63, August 1984.
- Mishra, M., Das, P.K., and Sarangi, S., 2004, "Transient Behaviour of Crossflow Heat Exchangers with Longitudinal Conduction and Axial Dispersion", J. of Heat Transfer, Vol. 126, No. 3, pp. 425-433.
- Pescod, D., 1979, "A Heat Exchanger for Energy Saving in an Air-Conditioning Plant", ASHRAE Trans., Vol. 85, No. 2, pp. 238-251.

- Pescod, D., 1968, "Unit Air Cooler Using Plastic Heat Exchanger with Evaporatively Cooled Plates", Australian Refrigeration, Air-conditioning and Heating, V.22, pp.22, September 1968.
- Pescod, D., 1974, "An Evaporative Air Cooler Using a Plate Heat Exchanger", CSIRO Technical, Report No.TR2, Victoria, Australia
- Peterson, J.L., 1993, "An effectiveness model for indirect evaporative coolers", ASHRAE Transactions, Vol. 99, Pt.2, pp.392-399.
- Patankar, S.V., 1980, "Numerical Heat Transfer and Fluid Flow", McGraw-Hill, New York
- Said, W.K., and Khassaf, F.H., 1984, "Indirect Evaporative Cooling for Residential Comfort", Al-Muhandis, No. 85, PP. 36-46, Des. 1984.
- Wooldridge, M.J., Chapman, H.L., and Pescod, D., 1976, "Indirect Evaporative Cooling Systems", ASHRAE Trans. pp. 146-153.
- Yellott, J.I., and Gamero, J., 1984, "Indirect Evaporative Air Coolers for Hot, Dry Climates", ASHRAE Trans., Vol.90, Pt.1B, pp.139-147.

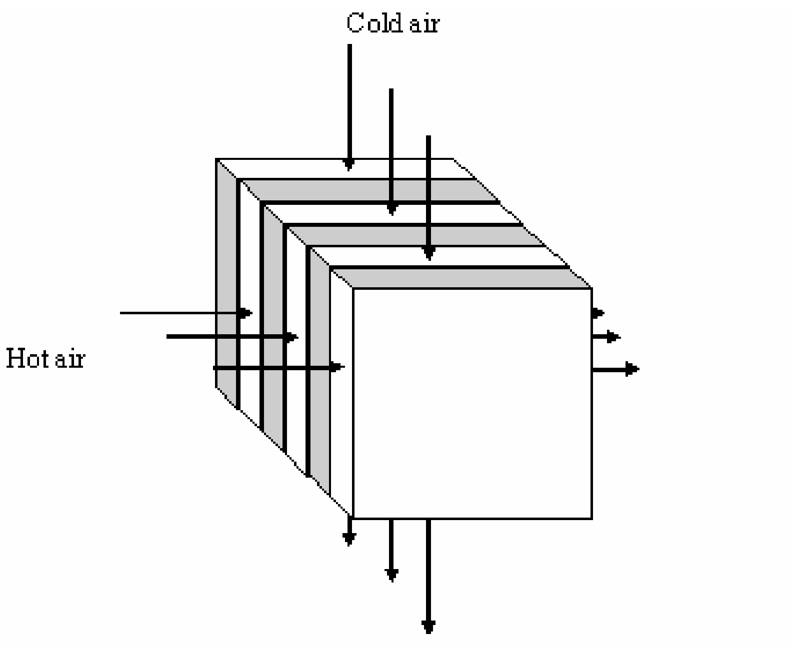
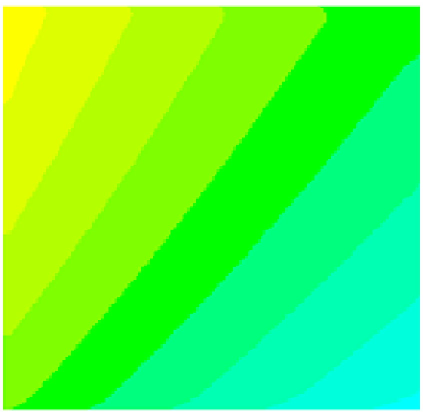
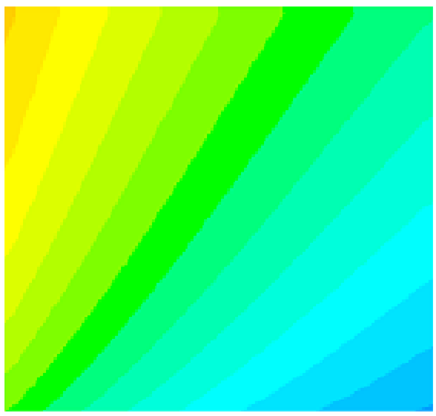


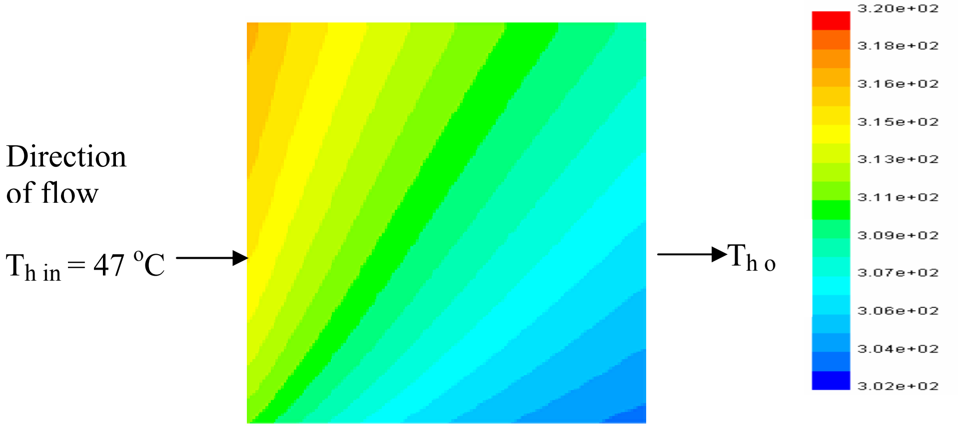
Figure. 1. Crossflow plate heat exchanger



a) 300X300 mm

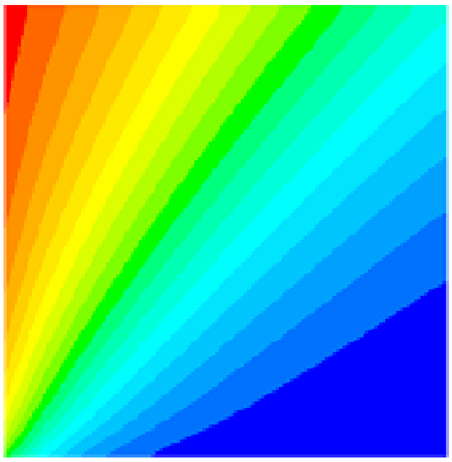


c) 700X700 mm

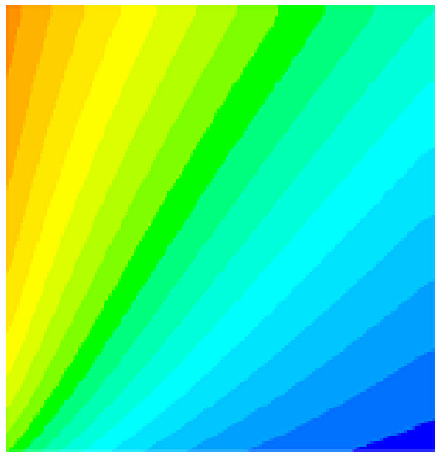


e) 1000X1000 mm

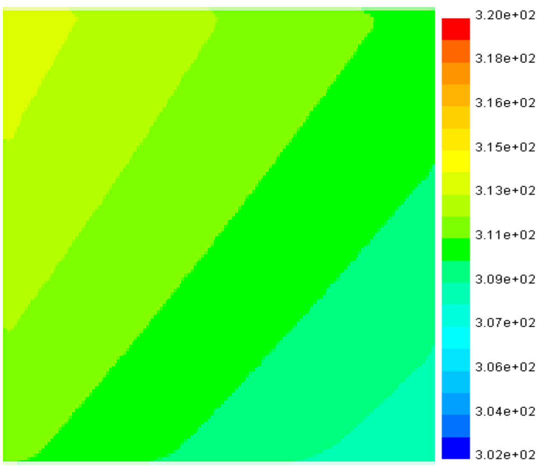
Figure. 2. The effect of PHE dimensions on hot air temperature distribution for 5 mm spacing and 2 m/s inlet air velocity



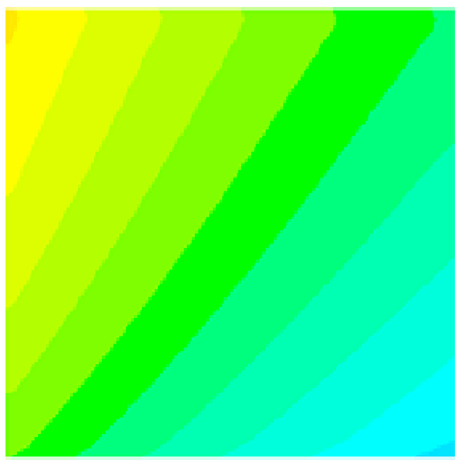
a) Spacing = 3 mm



b) Spacing = 5 mm



c) Spacing = 10 mm



d) Spacing = 15 mm

Figure. 3. The effect of plate spacing on the hot air temperature distribution for an 800X800 mm PHE at inlet air velocity of 1 m/s

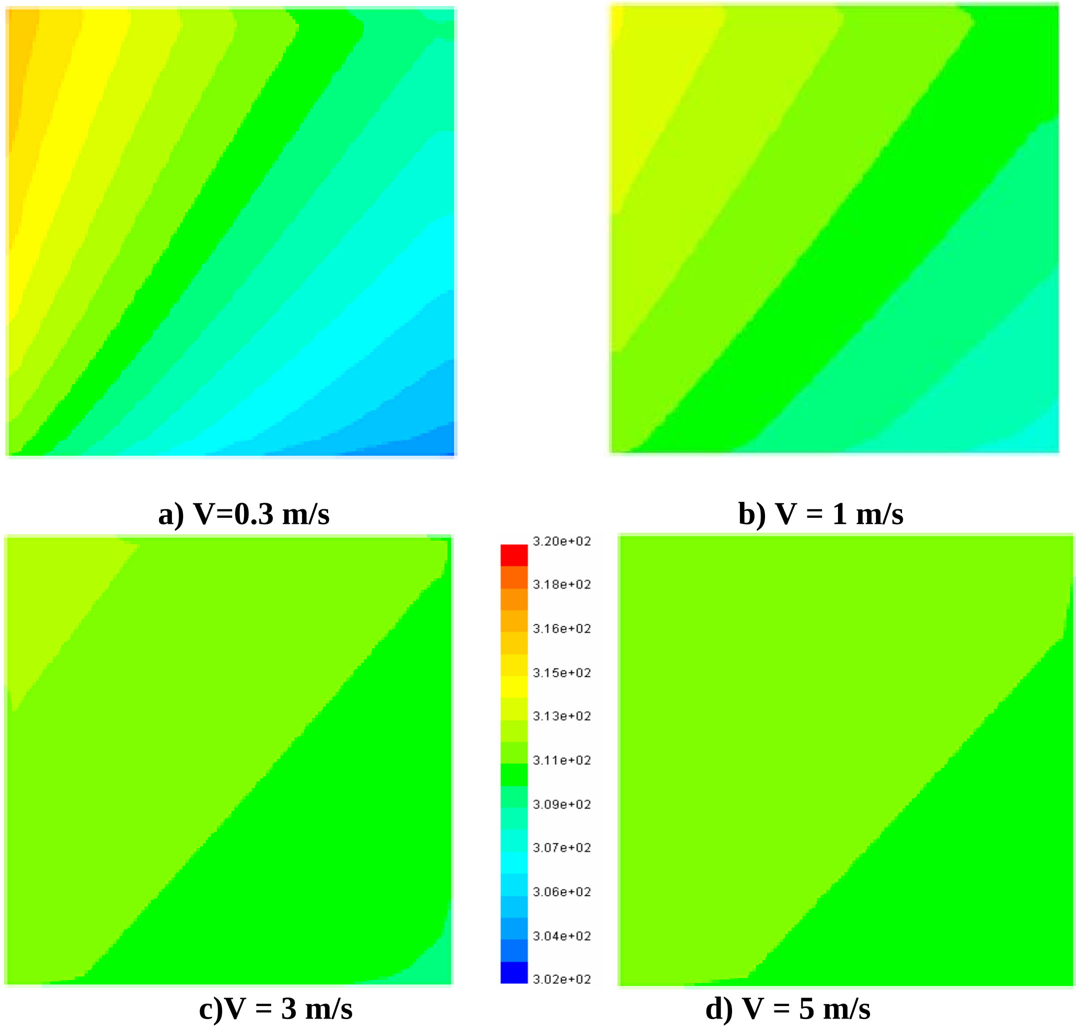


Figure. 4. The effect of inlet air velocity on hot air temperature distribution for the 300X300 mm PHE and 8 mm plate spacing

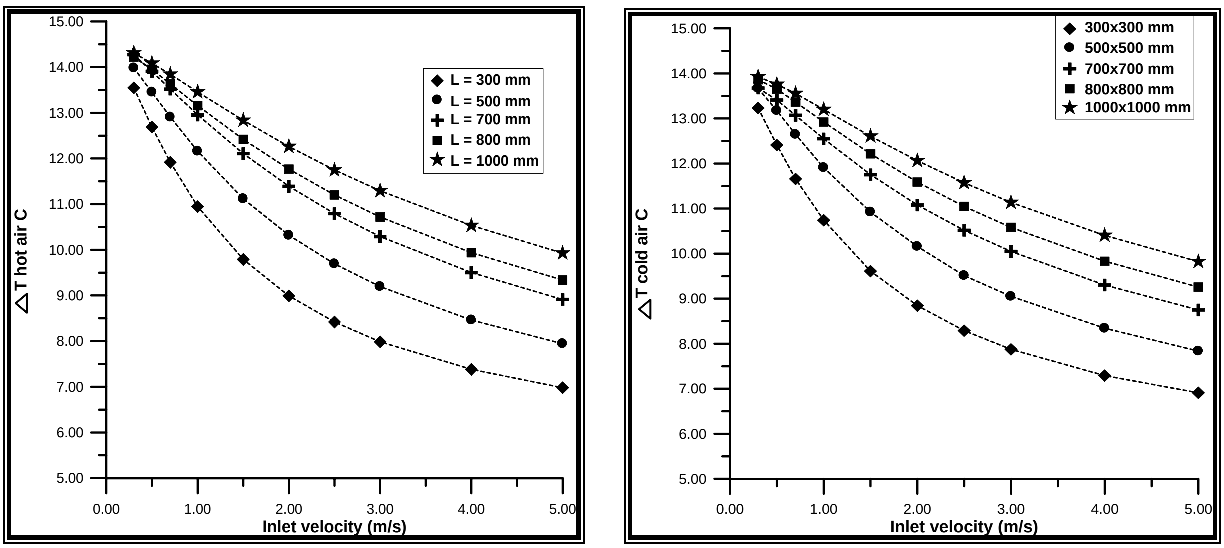


Figure. 5. The temperature difference for the two air streams at 5 mm spacing

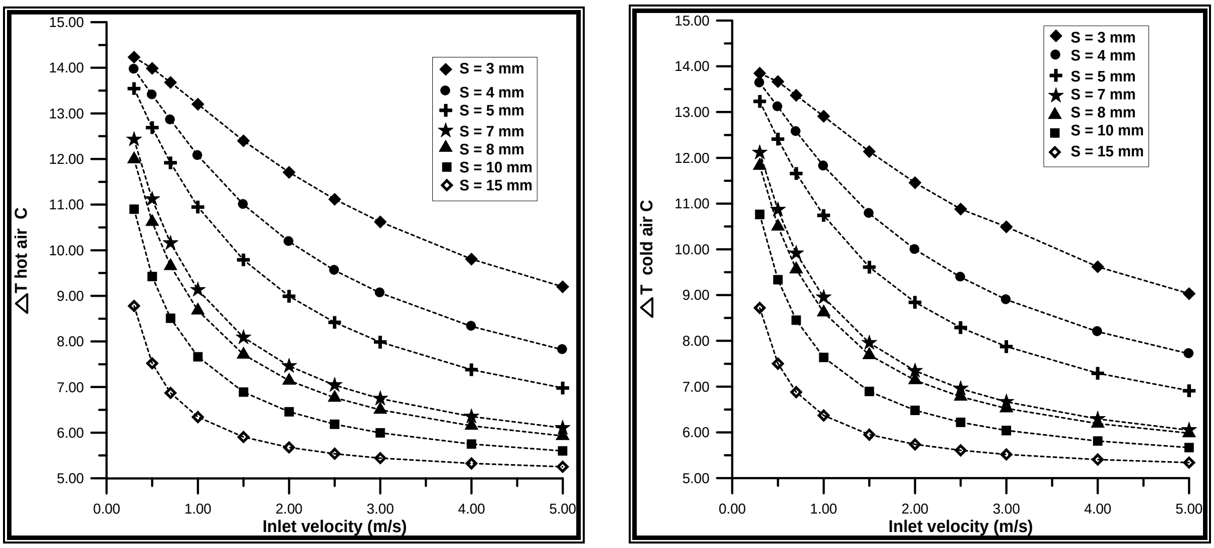


Figure. 6. The temperature difference for the two air streams at 300x300 mm PHE dimension

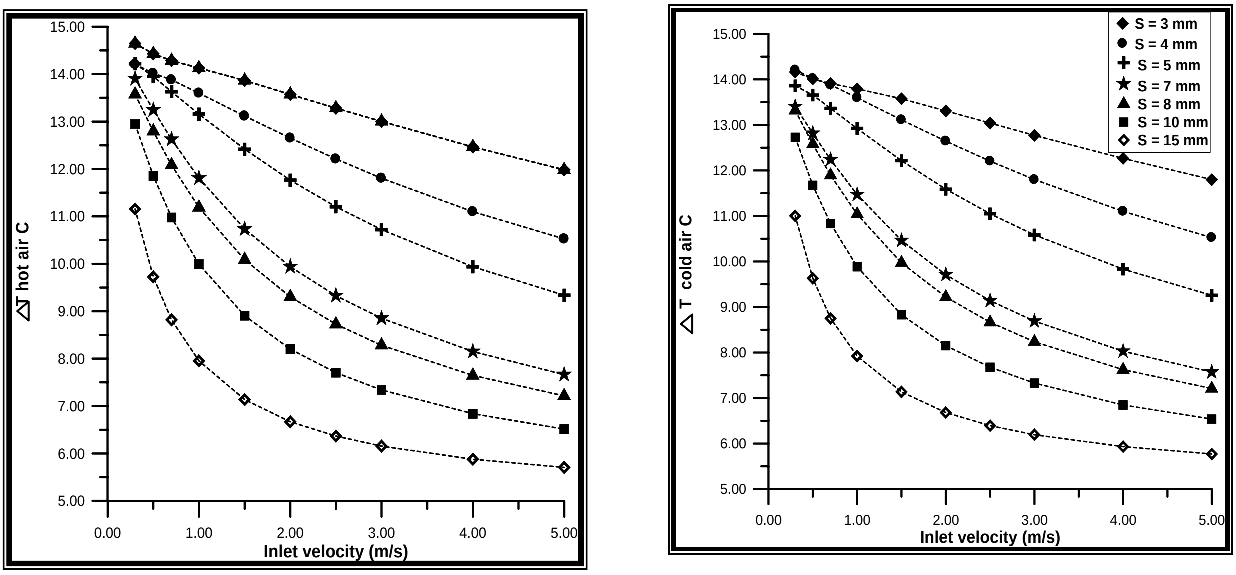


Figure. 7. The temperature difference for the two air streams at 800x800 mm PHE dimension

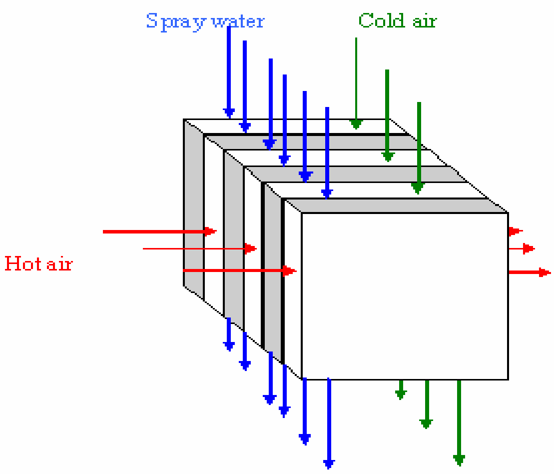


Figure. 8. Evaporatively plate heat exchanger

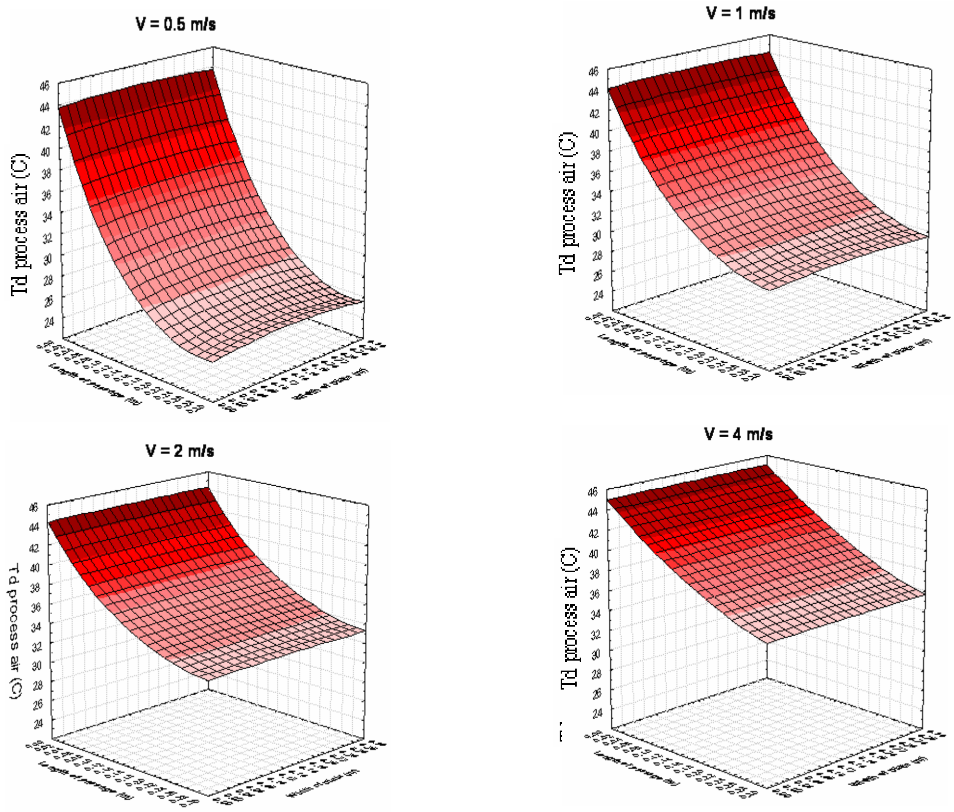


Figure. 9. The effect of inlet air velocity on the process air temperature distribution for the dimensions 300x300 mm and 8 mm plate spacing

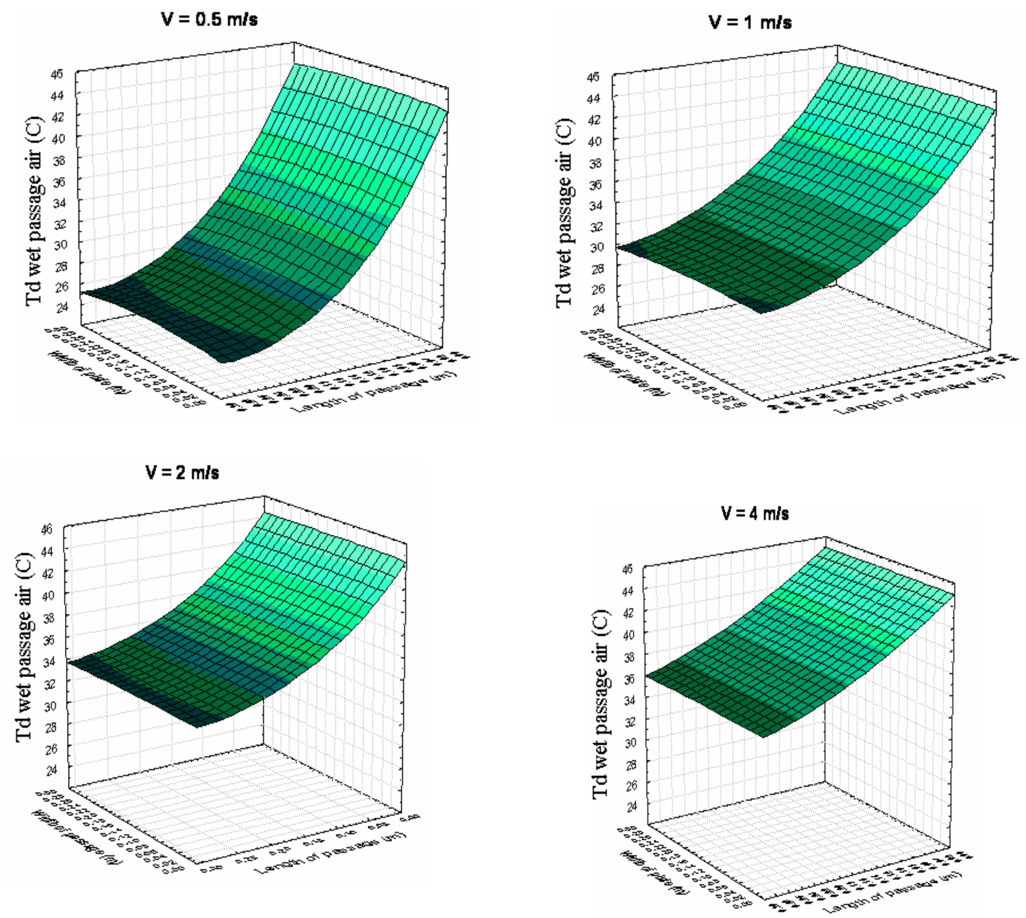


Figure. 10. The effect of inlet air velocity on the distribution of the wet passage air temperature for the dimensions 300x300 mm and 8 mm plate spacing

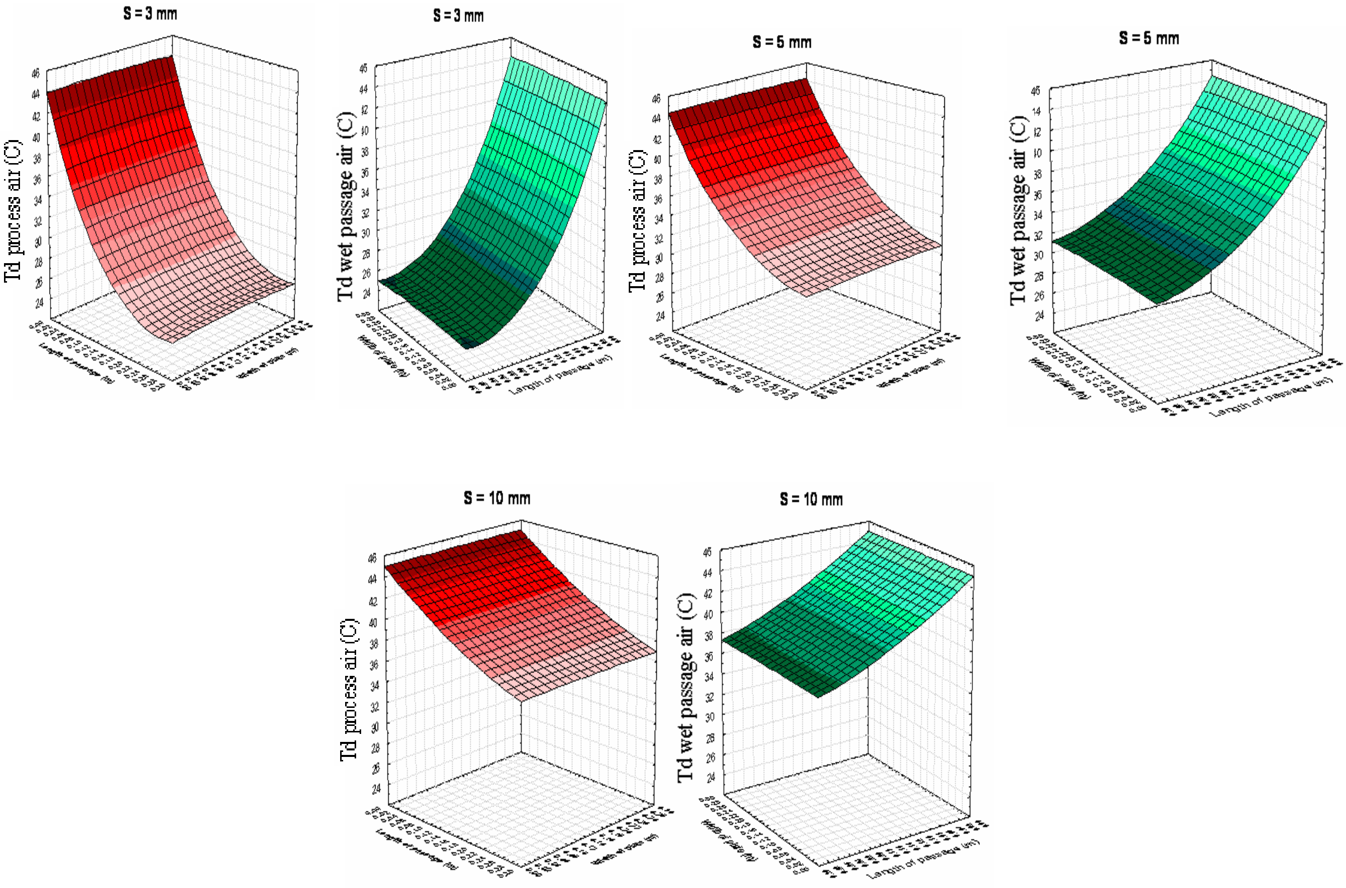


Figure. 11. The effect of plate spacing on the process air and the wet passage air temperature distribution for the dimensions 300x300 mm and 3 m/s inlet air velocity

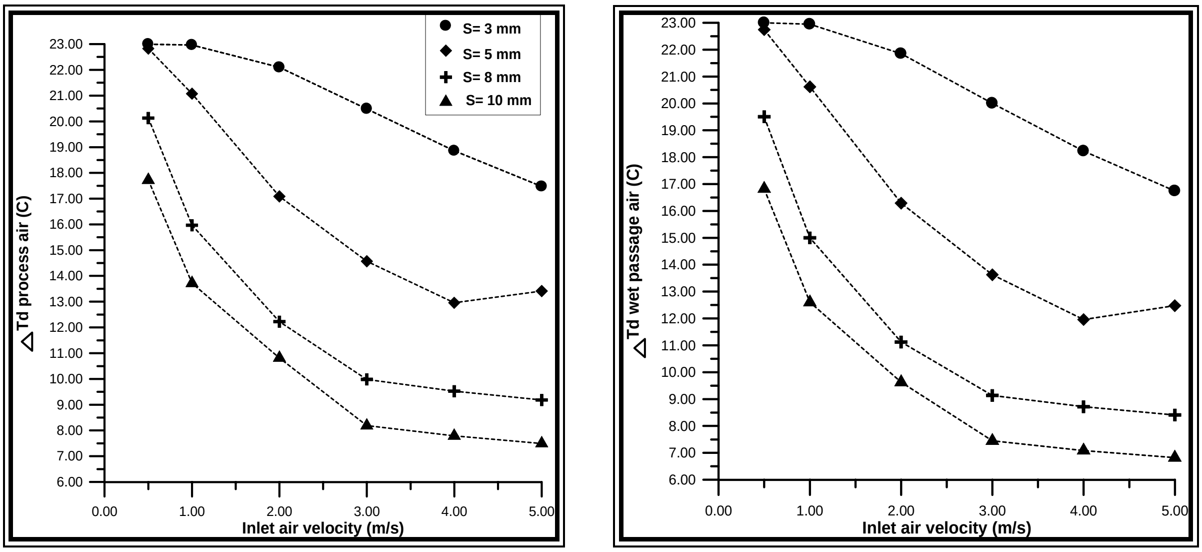


Figure. 12. The dry bulb temperature drop for the process air stream and for the wet passage air for 300x300 mm dimensions

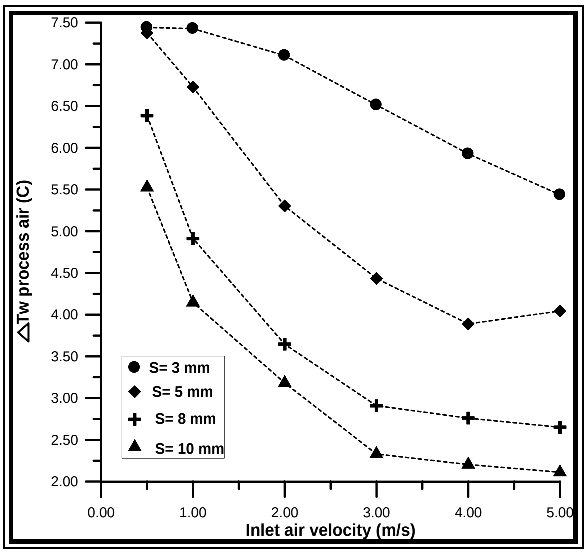


Figure. 13. The wet bulb temperature drop in the process air for 300x300 mm dimension

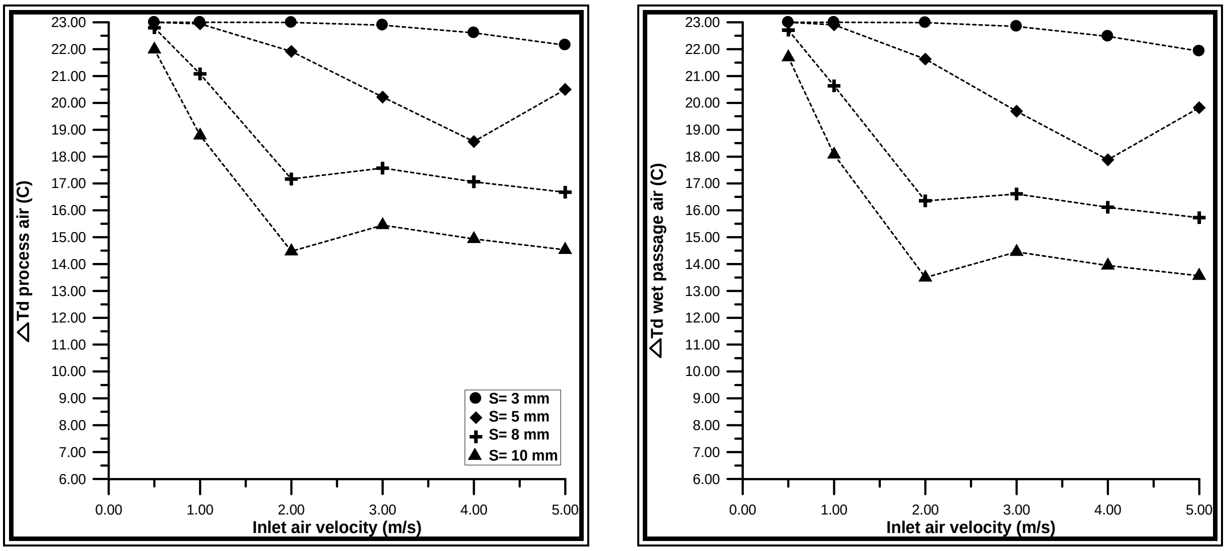


Figure. 14. The dry bulb temperature drop for the process air stream and for the wet passage air for 800x800 mm dimensions

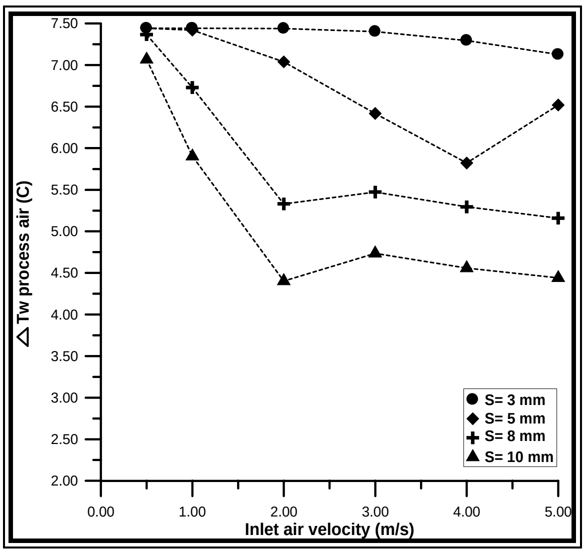


Figure. 15. The wet bulb temperature drop in the process air for 800x800 mm dimensions

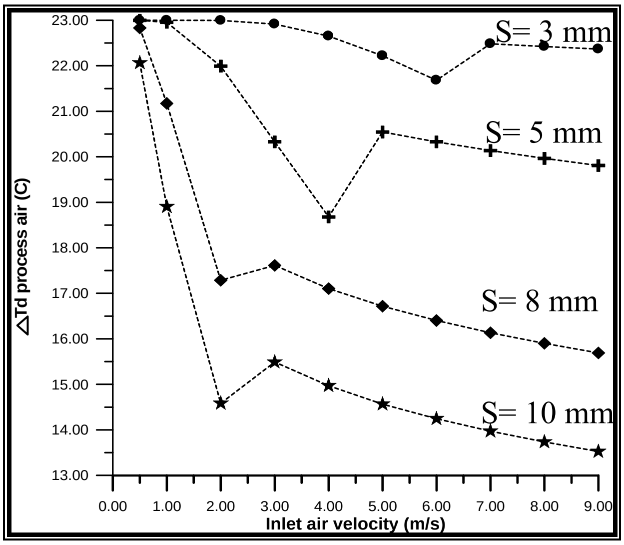


Figure. 16. Occurrence of turbulence at different plate spacing

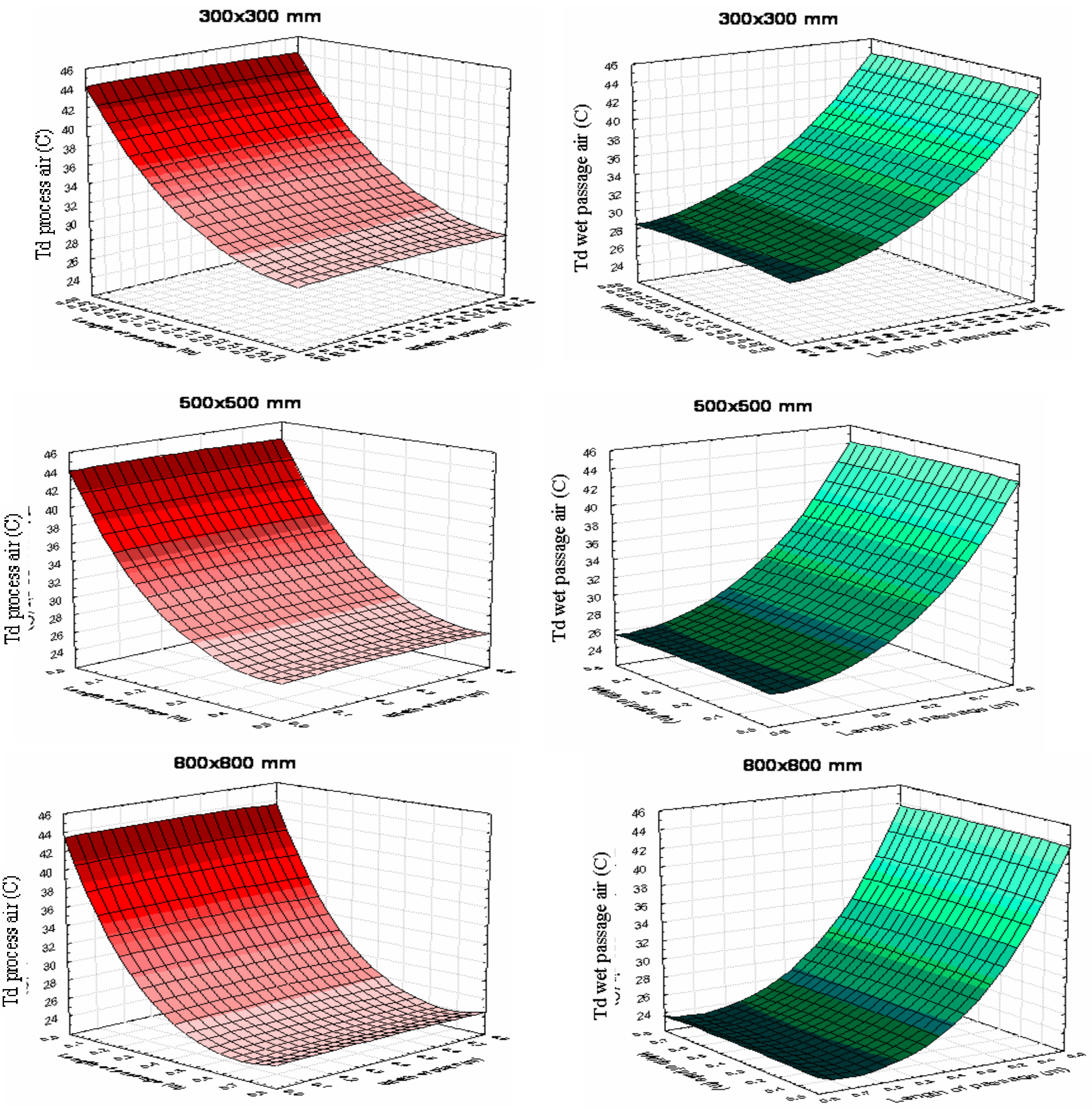
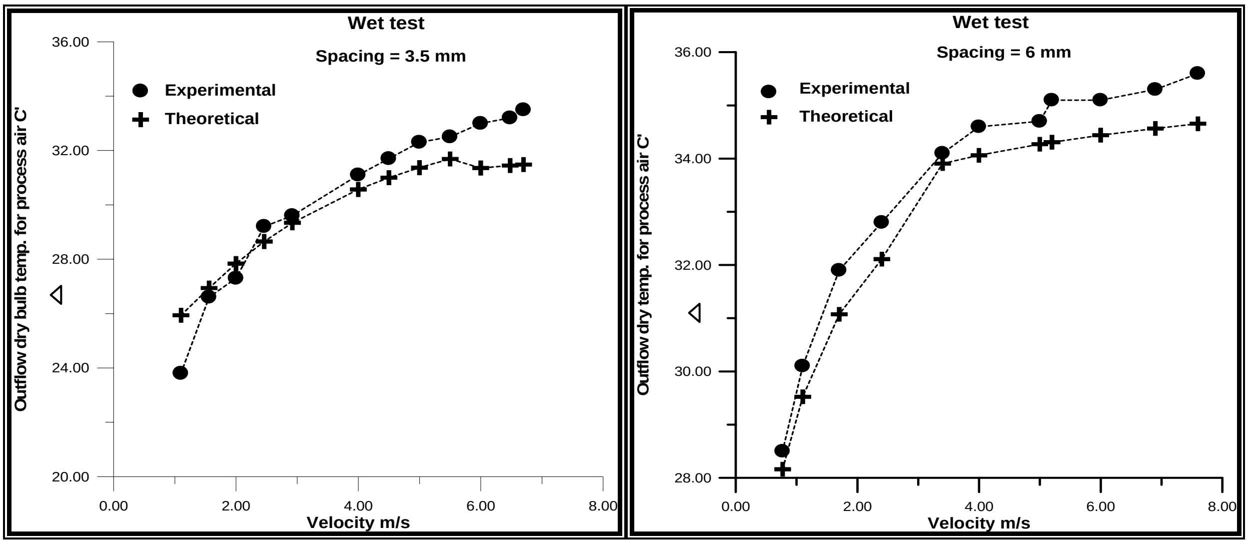


Figure. 17. The effect of various heat exchanger dimensions on the process air and the wet passage air temperature distribution for 5 mm plate spacing at 2 m/s inlet air velocity



(A) 3.5 mm plate spacing **(B) 6 mm plate spacing**

Figure. 18. Comparison between present theoretical results with experimental results of Enwia (1986)

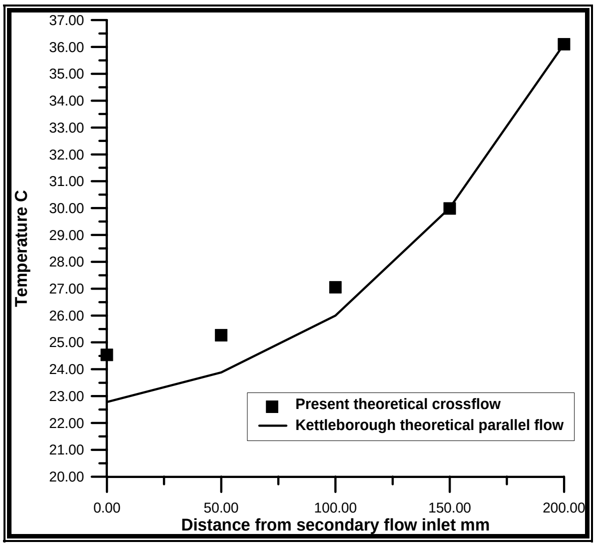


Figure. 19. Comparison of wet PHE between theoretical work for parallel flow by Kettleborough and Hsieh 1983 and the present theoretical work for crossflow PHE

NOMENCLATURE

Symbol	Description	units
A	Heat transfer area	m^2
a	Plate spacing in a heat exchanger	m
b	Width of plate	m
c_p	Specific heat at constant pressure	kJ/(kg.K)
c_{pu}	Specific heat for humid air	kJ/(kg.K)
D	Diameter	m
D_{ab}	Diffusion coefficient between two substances	m^2/s
d_h	Hydraulic diameter	m
H	Specific enthalpy	kJ/kg
h	Heat-transfer coefficient;	$W/(m^2.K)$

Symbol	Description	units
h_{fg}	Latent heat of evaporation	kJ/kg
k	Thermal conductivity	W/ (m.K)
k_m	Mass transfer coefficient	m/s
L	Length	m
Le	Lewis number $= h_i / k_m c_{pu}$	-
$\dot{m}$	Mass-flow rate	kg/s
Nu	Nusselt number = $h d_h / k$	-
P	Dimensionless pressure	-
p	Pressure	N/m^2
Pe	Peclet number = Pr * Re	-
Pr	Prandtl number = $c_p \mu / k$	-
q	Heat	kJ
Re	Reynolds number $= \rho v d_h / \mu$	-
s	Plate thickness	m
Sc	Schmidt number = υ / D_{ab}	-
Sh	Sherwood number = $k_m d_h / D_{ab}$	-
T	Temperature	K, ^oC
U	Dimensionless velocity component in the x-direction	-
U_o	Overall heat transfer coefficient	W/(m^2.K)
u	Velocity component in the x-direction	m/s
V	Dimensionless velocity component in the y-direction	-
v	Velocity component in the y-direction	m/s
w	Moisture content	kg/kg$_{d.a}$
X	Nondimensional distance in x-coordinate (x/L)	-
Y	Nondimensional distance in y-coordinate (y/L)	-

Greek symbols

ρ	Fluid density	kg/m^3
μ	Dynamic viscosity	kg/(s.m)
υ	Kinematic viscosity	m^2/s

Subscripts

c	cold fluid
ci	cold air-water interface region
d	dry-bulb
h	hot fluid
i	interface
in	inlet
l	latent
s	saturated; sensible; spray water temperature
w	wet-bulb