An Investigation of Indirect Evaporative Coolers, IEC With Respect to Thermal Comfort Criteria

M. Shariaty-Niassar[*], N. Gilani

Transport Phenomena & Nanotech. Lab, Chem. Eng. Faculty;
University College of Eng.,University of Tehran. Iran

Abstract
In this work, the effects of air stream direction in the channels of indirect evaporative cooler (IEC) on system performance have been investigated. In addition, the dependence of system performance on outdoor air temperature and relative humidity has been studied to determine the allowable conditions for proper operation of the system, with respect to thermal comfort criteria. For this; the different types of IECs were investigated using the CFD technique. Several codes were defined in MATLAB for modeling the parallel flow, counter flow and cross flow layout. The CFD program was validated against theoretical data from the literature and good agreement between the prediction and measurement was achieved. The calculated results show that when the air relative humidity is lower than 70%, the system can prepare a good indoor condition even at 50^oC, and a higher performance is achieved by using the IEC with counter current configuration. The results showed that IECs can be successfully used in hot and humid climates to fulfill the indoor thermal comfort conditions.

Keywords: *Modeling, Indirect evaporating coolers, Thermal comfort, Performance*

1. Introduction
The cooling energy demand world wide has increased tremendously in recent decades. This has created serious concern for which, in some countries, further utilities and hence additional supply had to be taken into account, thus increasing the average cost of electricity. Of course, this increase of energy consumption has environmental side-effects related to the increased CO_2 emissions and to the ozone-depleting ChloroFluro Carbons

(CFCs) used in air conditioners. The Kyoto protocol binds the developed countries to reduce the collective emissions of six key greenhouse gases - among which CO_2- at least by 5 % by 2008-2012. This protocol encourages the governments, amongst others, to improve energy efficiency and to promote renewable energy (EU, 2003). Therefore, counterbalancing the energy and environ-mental effects of air conditioning is a strong requirement for the future. Lately, research

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[*] Corresponding author: mshariat@ut.ac.ir

has been oriented towards low-energy techniques, one of which is evaporative cooling technique.

The main advantages of such systems are its simplicity, high cooling, low operational and maintenance costs, saving of fossil fuels and related emissions. Evaporative cooling is evaporation of other fluids in the presence of a draught, with a consequent cooling of the air. Evaporative cooling is especially well suited where the air is hot and humidity is low. However, in higher humidity areas there are many proven cost effective applications for evaporative cooling that makes it the right choice, for example, industrial plants, commercial kitchens, laundries, greenhouses, spot cooling (such as loading docks, warehouses, factories, construction sites, athletic events, workshops, garages) and indoor farming (poultry ranches, diary etc).

Evaporative cooling occurs when the vapor pressure of water is higher than the corresponding partial vapor pressure in the adjacent air. Such systems have a great potential to provide thermal comfort in places where the wet bulb temperature is low. Evaporative cooling equipment can be of the direct evaporative cooling (DEC) type or indirect evaporative cooling (IEC) type. In a DEC, water is vaporized in the airstreams and the heat and mass transfer between air and water cause the air dry bulb temperature to decrease and its humidity to increase, while the enthalpy is kept constant. However the minimum air temperature is the wet bulb temperature (WBT).

If not properly designed DECs may cause the following problems:

- The cooled air may be excessively humid and create thermal discomfort.
- The high rate of air flow and large number of air change cause an increase in wasted energy, which is required to cool discharged air.

In an IEC, the primary air stream which is used to provide the cooling load of a building, is separated from a wetted surface by a flat plate or tube wall as shown in Fig.1. The secondary air stream flows over the wetted surface and exhausts to ambient air. The liquid water is evaporated to the secondary air stream and extracts heat from the primary air stream through the flat plate or tube wall. Thus, the moisture content of air which flows through the primary channel remains constant. Finally, the primary cool air is used for cooling and ventilation.

There are some advantages to IECs which can be summarized as shown below:

- Depending on the system performance, the operating cost decreases from 20% to 60% below that of refrigerant air conditioning. Of course, the temperatures achieved by IECs are higher and variable, unlike the conventional systems.
- Power consumption is lower compared to that of other options. That is why IECs can also be used where electricity is expensive or less accessible.
- IECs can be used as a pre-cooler for refrigerant air conditioning systems.

Various investigations have studied the relative merits of IECs. Maclaine-Cross and Banks [1] developed- by analogy from published solutions for dry bulb temperature in dry surface heat exchangers- an analytical solution to the indirect evaporative cooling processes based on the following assumptions: moisture contents of air, in equilibrium with water, a linear function of the water surface temperature; the evaporating water maintains stationary and continuous at the same temperature. Due to high values of water surface tension, the wall surface of cooling air passages cannot be completely wetted with spray water and this leads to a reduced mass transfer area for film evaporation. In order to improve the model accuracy, a wettability factor was utilized to describe the effect of incomplete wetting conditions. Kettleborough and Hsieh [2] used a simple wettability factor to describe the effect of incomplete wetting. The changes
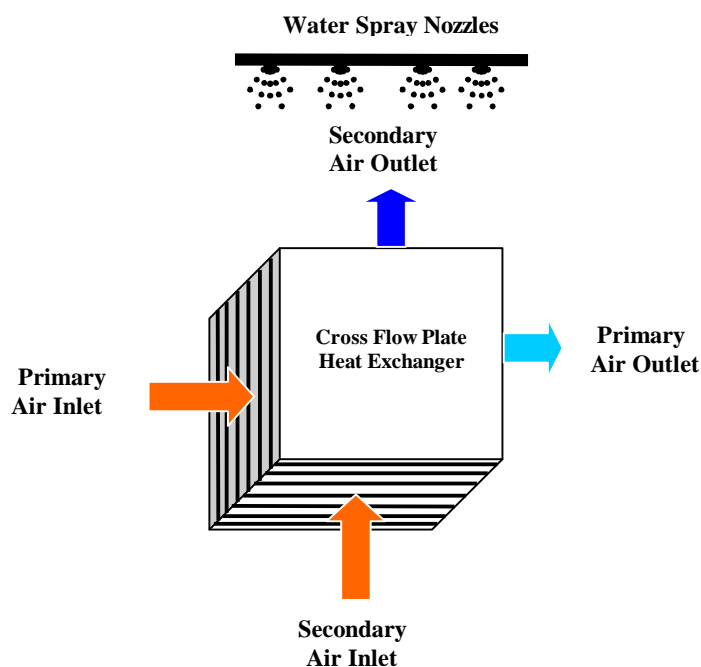
Water Spray Nozzles

Secondary Air Outlet

Cross Flow Plate Heat Exchanger

Primary Air Inlet

Primary Air Outlet

Secondary Air Inlet

Figure 1. Schematic diagram of indirect evaporative cooler.

of the spray water temperature along the heat exchanger surface were also taken into consideration. Numerical analysis was utilized to study the thermal performances of a counter flow indirect evaporative cooler unit. The agreement between the calculated and measured performance data was improved qualitatively. Erens and Dreyer [3] discussed three analytical models. Sample calculations showed that the optimum shape of the cooler unit would result in a primary to secondary air velocity ratio of about1.4, assuming that the primary and the secondary air mass flow rates are the same and that the same plate spacing were used. Navon and Arkin [4] were concerned with the possible utilization of a direct-indirect evaporative cooler in residences. It is shown that such a system can provide a significantly higher level of thermal comfort. Klitsikas *et al.* [5] investigated the performance of an indirect evaporative cooler by simulating its operation in a typical

building, using the weather conditions of Athens. Simulations were performed with the TRNSYS code and the results showed that the cooler increases the number of comfort hours per day inside a building. It was also found that night ventilation increases the comfort hours of the building while reducing the hours of operation of the cooler. Stoichkov and Dimitrov [6] introduced a correction factor of IEC for cross flow plate heat exchanger. Guo and Zhao [7] analyzed the thermal performance of an indirect evaporative air cooler numerically. The results showed that the coefficient of performance tends to be very high because the system consumes only fan and water pumping power. Joudi and Mehdi [8] investigated the application of the IEC to provide the variable cooling load of a typical dwelling at Iraq. The results showed that IEC can provide indoor good condition for most periods of the system operation and the

performance coefficient is high because it consumes a small amount of electricity for fan and water pump. Ren and Yang [9] developed an analytical model for the coupled heat and mass transfer processes under real operating conditions with parallel counter-flow configurations. Conventionally, one-dimensional differential equations were used to describe the heat and mass transfer processes. In modeling, values of Lewis number and surface wettability were not necessarily specified as unity. Effects of spray water evaporation, spray water temperature variation and spray water enthalpy change along the heat exchanger surface were also considered in model equations. Hettiarachchi *et al.* [10] investigated the effect of the longitudinal heat conduction in the exchanger wall of a compact-plate cross flow IEC with the NTU method numerically. The results of this research showed that the thermal performance deterioration of evaporative coolers may become significant for some typical operating conditions and could be as high as 10%, while it lies at less than 5% for most conservative conditions. Martin [11] studied heat recovery with a semi-indirect evaporative cooler by the experimental design method. The results showed that for low relative humidity contents and high temperatures of air supply, the main effect is evaporative from the surface of the ceramic pipes. For high temperatures and relative humidity of the air supply, dehumidification takes place and thus condensation appears in the exterior surface of the pipes and the latent and sensible heat recovered are added. A possible use of this recovery system can be in climates with high temperatures and humidity, such as tropical environments where the system could reduce the humidity of the primary air supply by using the cooling power of the secondary air.

Most researchers have investigated the effects of physical and geometrical parameters on system performance and some introduced better models. In the present paper, attempts have been made to investigate the effect of air flow direction in primary and secondary channels on the thermal performance of IEC. In addition, the conditions for the best operation point of indirect evaporative cooler, with respect to thermal comfort conditions, have been determined. For this, two simplified steady state models are developed to determine the air mean temperature at the outlet of the IEC primary channel, the first model for cocurent and countercurrent, and the second for cross flow configuration. In modeling, heat and mass transfer from the water film into the air flow and the overall energy balances equations are taken into account.

2. Thermal comfort

Thermal comfort is essentially a subjective response, or state of mind, where a person expresses satisfaction with the thermal environment. While it may be partially influenced by a variety of contextual and cultural factors, a person's sense of thermal comfort is primarily a result of the body's heat exchange with the environment. This is influenced by four parameters: environmental parameters (air temperature, radiant temperature, humidity and air speed), and two personal parameters (clothing and activity level, or metabolic rate). Current comfort standards such as ANSI/ASHRAE 55-92 [12] and ISO/EN 7730 [13] are based on a more or less static model (Figs. 2 and 3). The physiological and psychological response to the thermal environment is basically the same throughout the year. These standards divide recommended comfort requirements into two categories: winter (heating season) and summer (cooling season). This approach can be used in less demanding applications for simpler implementation of air-conditioning algorithms.

In this paper, first, different types of IEC will be modelled and on the indoor thermal comfort equipment, the outdoor desired operating condition will be determined.

3. Evaporative cooling simulation and mathematical formulation

The modeling includes two models, one for simulation of parallel, cocurrent and countercurrent, and another for cross flow layout. In order to determine the thermal performance of different types, it is necessary to calculate the air temperature at the outlet of the primary channel. For this, heat and mass transfer from the water film into the air flow and the overall energy balances equations are taken into account. Some postulations are assumed to enable solving the mathematical model. The major assumptions are summarized as follows:

1. The heat exchange between the system and the surroundings is negligible.
2. The flows in the channels are laminar, and hydrodynamically and thermally are fully developed.
3. Radiation is negligible due to low temperature.
4. The interface temperature between water film and air is equal.
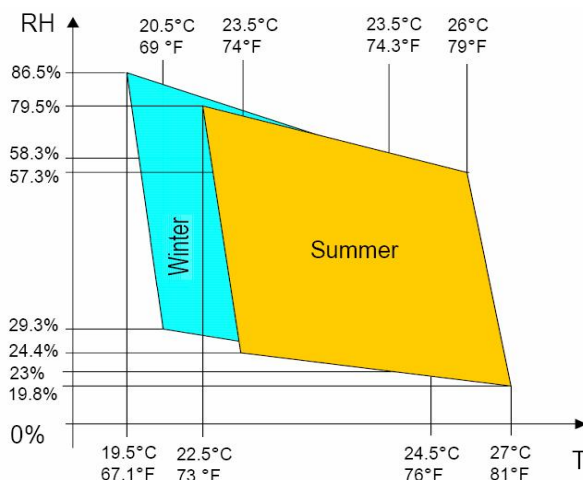5. The spray enthalpy is negligible.

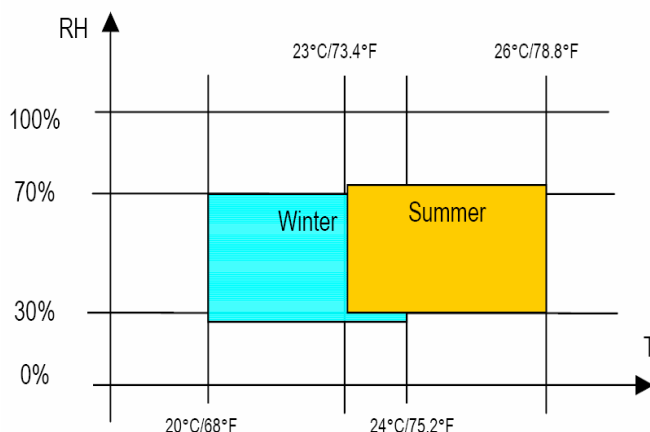**Figure 2.** Diagram showing the comfort zone according ASHRAE55 [12]

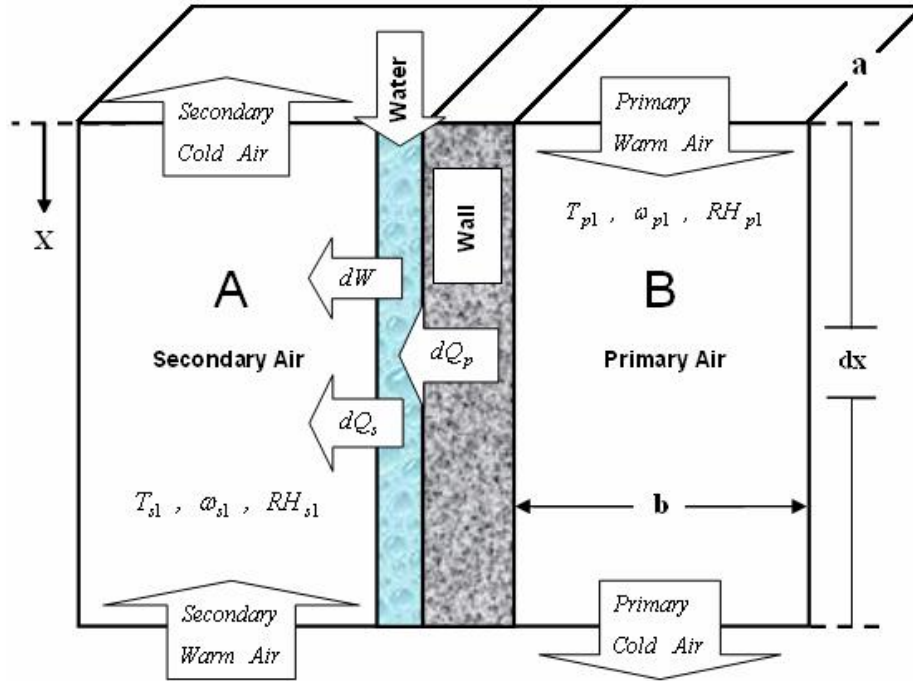**Figure 3.** Diagram showing the comfort zone according to ISO7730 [13].

Figure 4. Schematic Diagram of heat and mass transfer in an IEC.

6. The air enthalpy is expressed as a linear function of the wet bulb temperature only.
7. The Lewis number relating heat and mass transfer is taken as 1.0.
8. All thermophysical properties are evaluated at an average temperature.

3-1. Parallel, cocurrent and countercurrent IEC modeling

A schematic element of the basic model for an indirect cooler can be seen in Fig.4 in which the primary air is flowing in a downward direction, and the secondary air is flowing upward. The re-circulated water is sprayed onto the top of the heat exchanger and flows along the wall surfaces of cooling air passages as a thin film. Heat is then transferred from the film to the adjacent cooling air. At the interface between the air and the water spray, there is a film of saturated air in the close water. This film of saturated air is at the average water-temperature of the water spray film. Since the water-vapor partial-pressure at this interface

is higher than the vapor partial pressure in the air, there is a mass transfer of water-vapor into the air. This is associated with a latent heat transfer that is due to the water vaporization. At the same time, and due to the temperature difference between the surface of the water and the air, there is transfer of heat by convection.

3-1-1. Differential equations [14]

In principle and based on the energy and mass conservation law, a set of differential equations are to be considered along with the length of IEC as follows, according to the schematic diagram of heat and mass transfer in an IEC (Fig. 4).

• The heat transfer from the water film into the secondary air flow:

$$dQ_s = h_s \left(T_w - T_s \right) \, dA \tag{1}$$

• The mass flow of water that is evaporated into the secondary air:

$$dW = h_m \left(\omega_{(T_W)} - \omega \right) \, dA \tag{2}$$

- The heat transfer from the primary air into the water film:

$$dQ_p = U_z \left(T_p - T_w \right) \, dA \qquad (3.a)$$

$$dQ_p = -\overset{\bullet}{m}_s \, dE_p \qquad (3.b)$$

- The overall heat transfer coefficient is:

$$U_z = \frac{1}{\dfrac{1}{h_p} + \dfrac{\delta_{wall}}{k_{wall}} + \dfrac{1}{h_w}} \qquad (4)$$

- The water mass balance (refer to Fig.4), yields:

$$d \overset{\bullet}{m}_w = dW \qquad (5)$$

- The water and air mass balance (refer to Fig.4), yields:

$$\overset{\bullet}{m}_s \, d\omega = d \overset{\bullet}{m}_w \qquad (6)$$

- The overall energy balance on the process for the A and B control surface can be expressed as:

$$\overset{\bullet}{m}_s \, dH_s = -H_{pw} dW - dQ_s \qquad (7.a)$$

$$\overset{\bullet}{m}_s \, dH_s + dQ_p = \overset{\bullet}{m}_w \, dH_w + H_w d \overset{\bullet}{m}_w \qquad (7.b)$$

The enthalpy of humid air equals the sum of the enthalpies of the dry air and water vapor. The specific enthalpy of humid air is also defined per unit mass of dry air. For lower pressure, the specific enthalpy of water vapor is almost a linear function of temperature. Therefore, the enthalpy of humid air can be expressed as:

$$H(T) = C_p T + \omega \left(2.501 + 1.805 \times 10^{-3} T \right) \qquad (8)$$

By using equations (1)-(3) and rearrangement of equations (5)-(8), a set of ordinary

differential equations are described below [14]:

$$\frac{d\omega}{dx} = -\frac{h_m f_m a \left(\omega_{(T_w)} - \omega \right)}{\overset{\bullet}{m}_s} \qquad (9)$$

$$\frac{dT_s}{dx} = -\frac{a \left(T_w - T_s \right)}{\overset{\bullet}{m}_s c_p} \left(h_m f_m c_{pw} \left[\omega (T_w) - \omega \right] + h_s \right) \qquad (10)$$

$$\frac{dT_w}{dx} = \frac{a}{\overset{\bullet}{m}_w c_w} \left(h_m f_m \left[T_w (c_w - c_{pw}) - h_{fg} \right] \right.$$
$$\left. \cdot \left[\omega (T_w) - \omega \right] - h_s (T_w - T_s) + U_z (T_p - T_w) \right)$$

$$(11)$$

$$\frac{dT_p}{dx} = -\frac{U_z a \left(T_p - T_w \right)}{\overset{\bullet}{m}_p c_p} \qquad (12)$$

Equations 9-12 are the full description of the system. In the above equations, f_m represent wettability of the plate. The corresponding boundary conditions are introduced in Fig.5.

3-1-2. Governing heat and mass transfer coefficient equations

As far as cost, weight and corrosion resistance are concerned, Poly Vinyl Chloride (PVC) would be the ideal material for the plates. Although PVC has very low thermal conductivity, the thermal resistance of thin PVC plates becomes relatively small in comparison with the thermal resistance of the dry side and the air/water interface. The thermal resistance of the water film is negligibly small in comparison with the air/water interface thermal resistance. The mass transfer coefficient can be approximated using an analogy between heat and mass transfer [4]:

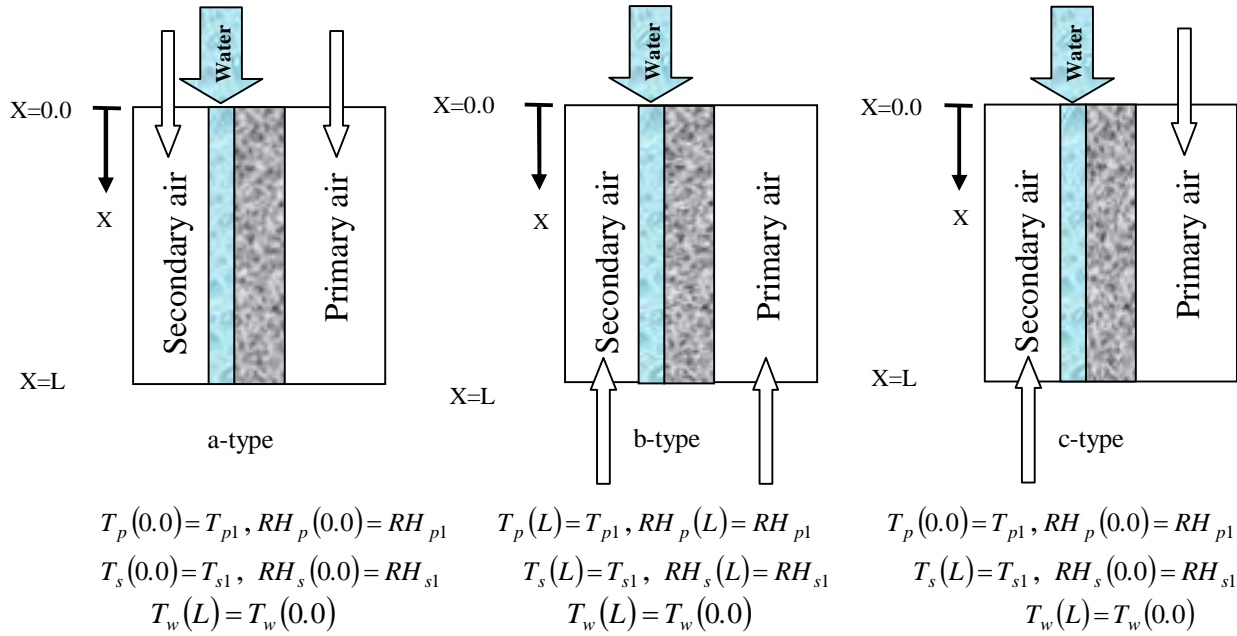$$h_m = \frac{h}{c_p Le} \qquad (13)$$

Figure 5. Boundary conditions

Where h, defined as the dry heat transfer between parallel plates is defined as [4] :

$$h = 54(u)^{0.7} \qquad (14)$$

For steady, fully developed, two-dimensional flow of a laminar film along a vertical surface, it follows that [3]:

$$u_w = 1.5\left(\frac{u_w g}{48}\right)^{1/3}(Re_w)^{2/3} \qquad (15)$$

$$\delta_w = \left(\frac{3u_w^2}{4g}\right)^{1/3}(Re_w)^{1/3} \qquad (16)$$

$$Re_w = \frac{\overset{\bullet}{m_w}}{L\mu_w} \qquad (17)$$

The water side film heat transfer coefficient can be expressed as [3]:

$$h_w = \frac{k_w}{\delta_w} \qquad (18)$$

3-2. Cross-flow IEC modeling

Cross-flow plate heat exchangers (CPHE) are widely used in air conditioning systems for energy recovery during the winter and cooling the outside air during the summer. In IEC, CPHE is also used. This case is very similar to the two cases introduced previously. Here, the flow directions in the primary and secondary channels are perpendicular as seen in Fig.6. There are several different models to express the heat and mass transfer process in IEC. Here, a simple and accurate model is introduced. An infinitesimal element for modeling of this type is seen in Fig.6. With assumptions that are expressed in section 3.1, a set of differential equations, by applying the basic energy and mass conservation law, are obtained as follows:

$$\frac{dT_p}{dx} = -\frac{U_z(T_p - T_w)}{\overset{\bullet}{m_p}c_p}dy \qquad (19)$$

$$\frac{d\omega}{dx} = -\frac{h_m f_m \, dy(\omega_{(T_w)} - \omega)}{\overset{\bullet}{m_s}} \qquad (20)$$
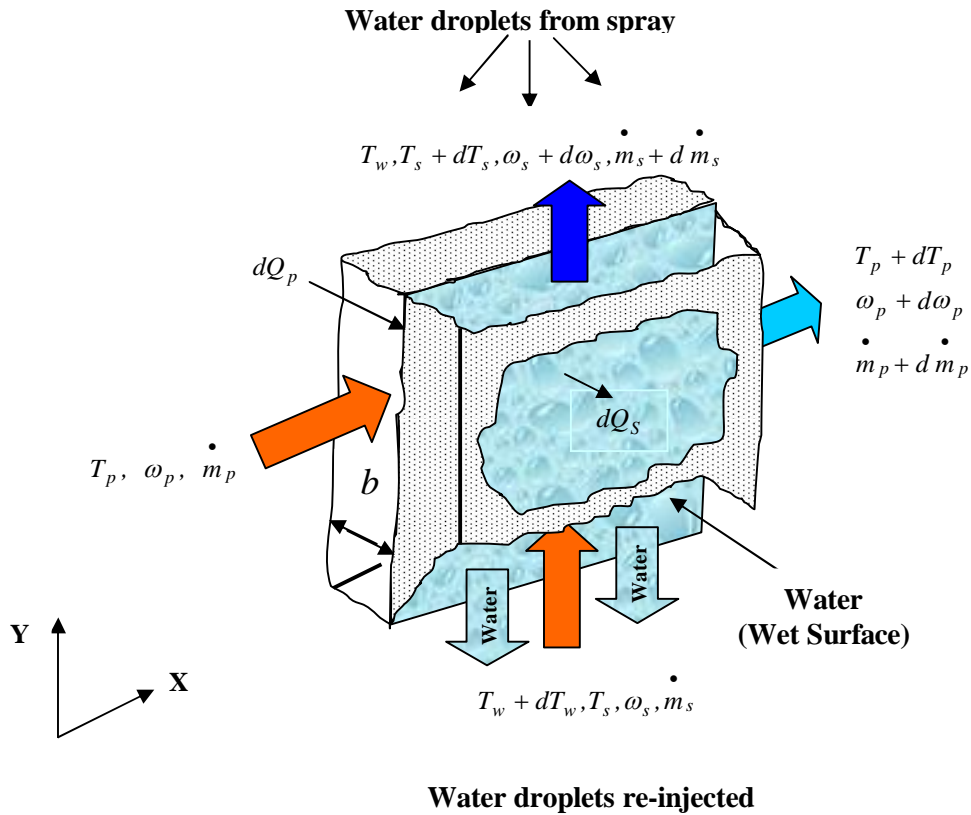
Water droplets from spray

Figure 6. Schematic Diagram of the heat and mass transfer in a crossflow IEC.

$$\frac{dT_w}{dy} = (\dot{m}_p c_p \frac{dT_p}{dy} + \dot{m}_s c_p \frac{dT_s}{dy} - \dot{m}_s c_p W T_w \frac{dW}{dy}) / \dot{m}_w c_{pw}$$

(21)

$$\frac{dT_s}{dy} = \frac{H_s(T_w) - H_s(T_s)}{c_p} \frac{dW}{dy} + \frac{h_s(T_w - T_s)}{\dot{m}_s c_p} dx$$

(22)

Where:

$$\dot{m}_p = \left(\frac{1}{2}\right) \rho u_p b dy \qquad (23)$$

$$\dot{m}_s = \left(\frac{1}{2}\right) \rho u_s b dx \qquad (24)$$

Equations (19-22) describe the operation of cross flow IEC. Boundary conditions for cross flow layout have been introduced in Fig.7.

4. Verification

There is no experimental data to validate the results of the theoretical model. So, the calculation has been carried out for cross flow type under the same conditions of the theoretical studies of Ref. [7] to check the mathematical model and to ensure the accuracy of the computations. Figs. 8, 9 and 10 show the theoretical results of the present model compared to the theoretical results of Ref. [7]. The quantitative comparison shows a reasonable agreement between the results obtained by the present study and the published results. It should be noted that the calculation were carried out at the same conditions of Ref. [7] in which plate spacing=5mm, $T_{p1}=33^oC$, $T_{s1}=25^oC$, $f_m=100\%$, $RH_{air}=70\%$, $u_p=3.0$m/s and $u_s=2.4$m/s.

For d-type:

$$T_p(0.0, y) = T_{p1}, \ RH_p(0.0, y) = RH_{p1}$$
$$T_s(x, 0.0) = T_{s1}, \ RH_s(x, 0.0) = RH_{s1}$$
$$T_w(x, L) = T_w(x, 0.0)$$

For e-type:

$$T_p(0.0, y) = T_{p1}, \ RH_p(0.0, y) = RH_{p1}$$
$$T_s(x, L) = T_{s1}, \ RH_s(x, L) = RH_{s1}$$
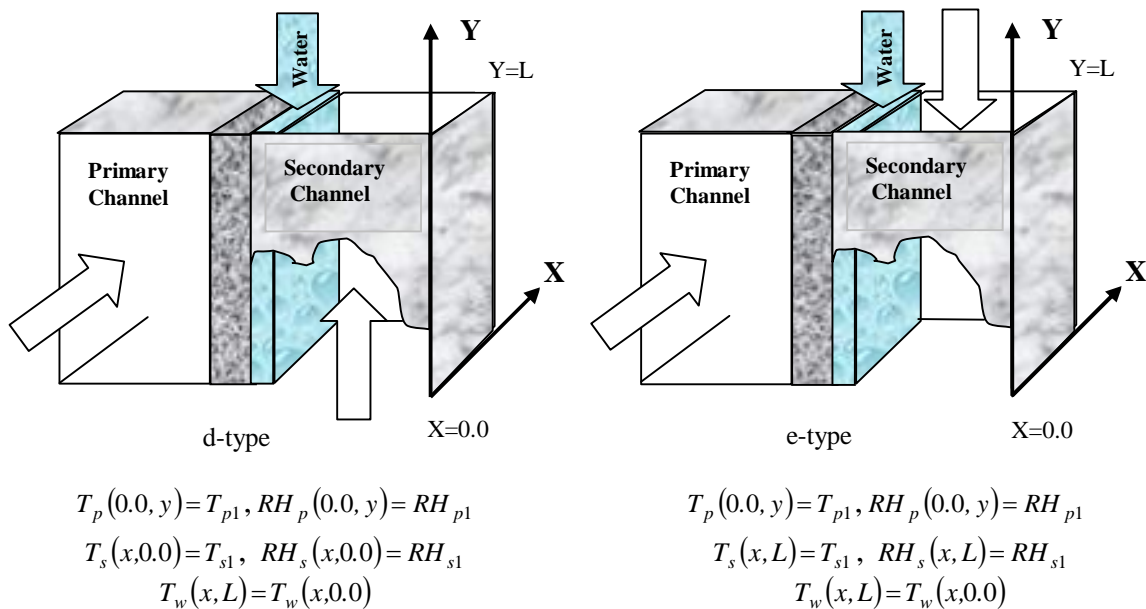$$T_w(x, L) = T_w(x, 0.0)$$

Figure 7. Boundary conditions.
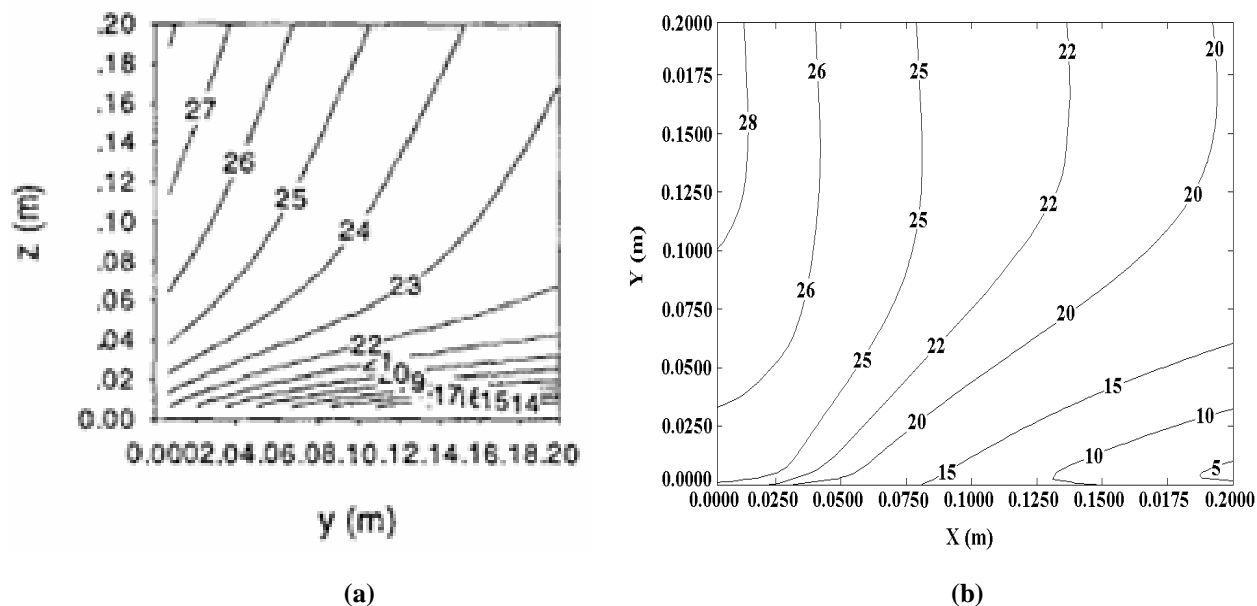

(a)

(b)

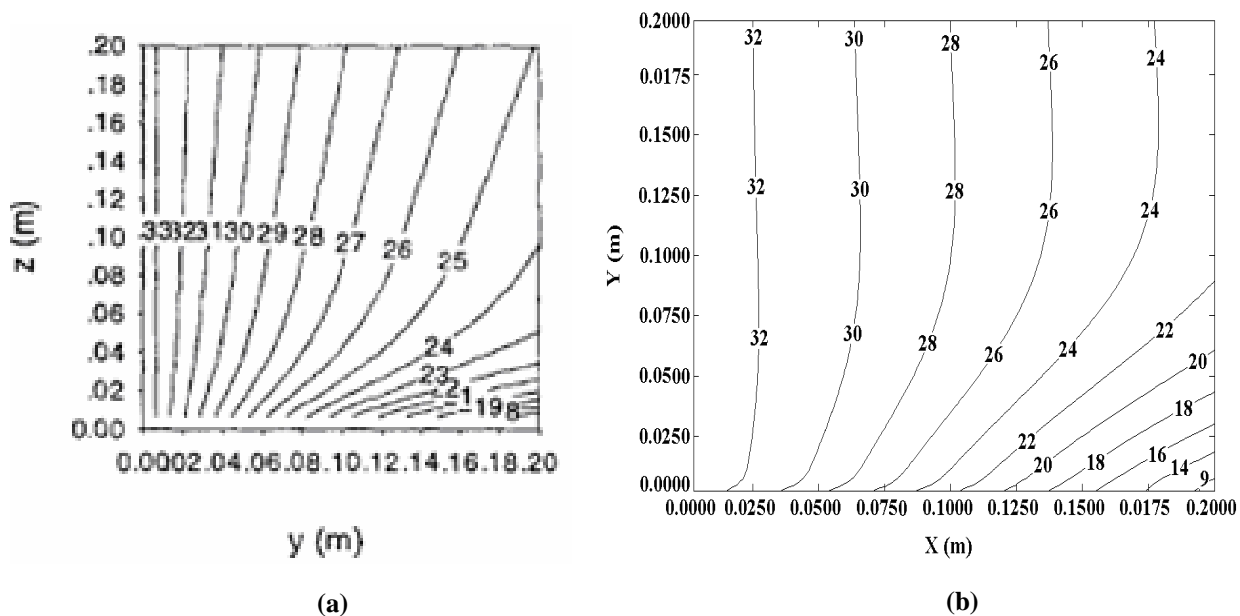**Figure 8.** (a): Water film temperature [7], (b): Water film temperature (present model)

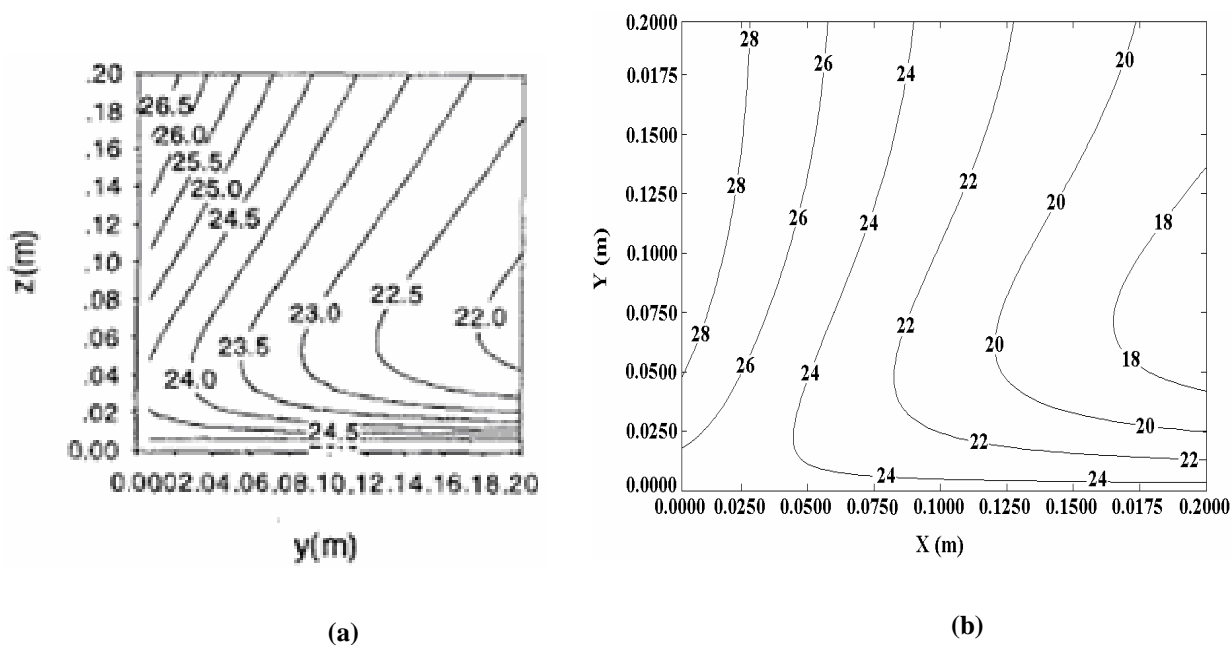**Figure 9.** (a): Primary air temperature [7], (b): primary air temperature (present model)

Figure 10. (a): Secondary air temperature[7], (b): Secondary air temperature (present model)

5. Numerical solution

The coupled governing equations 9-12 and 19-22 describe the heat and mass transfer process in IECs. The finite difference solution for these equations can be marched along the streams. The equations are discreted, then the upwind scheme is used to model the system. A computer program was written in MATLAB for solving the set of differential equations. In this program, the properties of air, water and air–water vapor mixture were assumed constant at each step of the numerical calculation. These equations were solved by the finite difference method based on the following procedures. A finer mesh size has been used to give an acceptable accuracy and a careful check for the grid-independence of the numerical solutions has been made to ensure the accuracy and validity of the numerical results.

(1) Initialize all the variables.
(2) Calculate heat transfer and mass transfer coefficients.
(3) Assume the water film temperature T_w and solve the governing differential equations, check the accuracy of the iterated value of T_w.
(4) Return to Step (3) and repeat the processes until convergence.

The cross-flow is slightly complicated because the water outlet temperature at the secondary air inlet side is not constant along the lower length of the cooler. Here, the water outlet temperature is assumed and the numerical solution process proceeds upwards until the inlet water temperatures are found for each element along the length of the cooler. The differences between the inlet and outlet water temperatures in each vertical column of elements are used to update the assumed water outlet temperature. The iterative process stops when the average inlet and outlet circulating water temperatures are the same and the calculated inlet water temperature (at the secondary air outlet side) is constant along the length of the cooler.

6. Results and discussion

The numerical solution for air within a range of 35-50°C with the relative humidity, R.H., of 30-70% is obtained. The system uses a 0.2m×0.2m PVC heat exchanger with a wettability ratio of 70%. The distance between the surfaces is taken as 10mm. The results of Table 1 show that when ambient R.H. is in the range of 30%‑ 70%, IEC is capable of supplying the desired internal air temperature. Comparison of the calculated results show that if the air relative humidity is in the range of 30-70% and air velocity in the channels is suitable, IECs can prepare a good indoor condition. The flow direction in the primary and secondary channels affects the performance and capability to produce cooled air too. The performance of direct and indirect evaporation cooling systems can be assessed on the saturation efficiency (SE), defined as:

$$SE = \frac{T_{db,1} - T_{db,2}}{T_{db,1} - T_{wb,1}} \tag{25}$$

The results of the numerical solution that have been reported in Table 1 indicate the distinction between the performances of IECs. In similar condition, the type which produces cool air in high velocity is superior because it produces a greater cooling load compared to other types. So, the saturation efficiency arrangement of IECs are as shown below:

$$SE_{c-type} > SE_{a-type} > SE_{d-type} = SE_{e-type} > SE_{b-type} \tag{26}$$

When the R.H. of the ambient is greater than 70%, the exit air humidity from the primary channel would stay out of the permitted range of the thermal comfort. This means that IECs are not suitable under these conditions. According to Table 1, when the ambient temperature is high, the air velocity in the IEC channel should be low to avoid thermal

discomfort. In low outdoor temperature and relative humidity, high air velocity in channels may cause thermal discomfort. In this condition, IEC can be used for a larger space.

Thus, in a small space, a low efficient or efficient type with big plate spacing must be used.

Table 1. Airstream velocity in IEC channels to produce inlet indoor good condition

IEC Type	Outdoor Air Relative Humidity	Outdoor Air Temperature(oC)				Primary Outlet Temperature (°C)
		35	**40**	**45**	**50**	
	RH	Primary air velocity (m/s)	Primary air velocity (m/s)	Primary air velocity (m/s)	Primary air velocity (m/s)	
a-type	30	5.5-7.5	4.0-4.9	3.15-3.7	2.6-3.0	
	40	4.8-6.4	3.4-4.2	2.6-3.15	2.15-2.5	
	50	4.0-5.3	2.85-3.45	2.2-2.6	1.75-2.1	
	60	3.0-4.2	2.2-2.8	1.7-2.05	1.35-1.6	
	70	2.3-3.3	1.06-2.05	1.2-1.5	0. 9-1.15	
b-type	30	3.2-4.7	2.1-2.8	1.45-1.9	1.1-1.4	
	40	2.6-4.0	1.65-2.3	1.15-1.5	0.85-1.1	
	50	2.0-3.1	1.3-1.8	0.85-1.15	0.65-0.8	
	60	1.0-1.75	0.95-1.38	0.6-0.85	0.4-0.6	
	70	1.35-1.75	0.62-0.9	0.32-0.58	0.2-0.3	
c-type	30	8.3-11.0	5.3-7.3	3.8-5.5	3.0-4.0	23.0-27.0
	40	7.0-9.5	4.9-6.3	3.8-4.5	2.7-3.7	
	50	5.7-7.7	4.0-5.0	3.0-3.7	2.5-3.0	
	60	4.4-6.2	3.0-4.0	2.3-2.8	1.8-2.2	
	70	3.2-4.7	2.1-2.8	1.6-2.0	1.2-1.5	
d-type	30	9.5-17.5	3.0-6.0	1.7-2.8	1.2-1.8	
	40	5.0-10.5	2.3-4.3	1.45-2.15	1.0-1.4	
	50	3.7-7.0	1.9-2.95	1.2-1.75	0.85-1.2	
	60	3.0-5.0	1.65-2.5	1.05-1.45	0.75-0.97	
	70	2.5-4.0	1.4-2.15	0.95-1.25	0.65-0.85	
e-type	30	9.5-17.5	3.0-6.0	1.7-2.8	1.2-1.8	
	40	5.0-10.5	2.3-4.3	1.45-2.15	1.0-1.4	
	50	3.5-6.9	1.9-2.95	1.2-1.75	0.85-1.2	
	60	2.8-5.1	1.65-2.5	1.05-1.45	0.75-0.97	
	70	2.3-4.0	1.4-2.15	0.95-1.25	0.65-0.85	

Cooling production in each primary channel of an IEC is given by:

$$Q_c = \dot{m}_p \, c_p \left(T_{p1} - T_{p2} \right)$$

(27)

Where T_{P1} and T_{P2} are inlet and outlet primary side temperature that specify outdoor and indoor comfort air temperature. Knowing the cooling demand of the building, the total numbers of primary channels are determined by dividing the cooling demand to Q_c.

7. Conclusions

Mathematical models for simulating IECs have been developed in this work. The results show the performance strongly depends on ambient air humidity and temperature. However, it is easy to prepare a good indoor thermal condition under lower humidity even at high ambient temperatures. It is found that that the IEC with proper configurations is capable of preparing a good indoor condition, even in high relative humidity (70%) and high ambient air temperature (50^{oC}). Therefore, this system is a suitable technique to supply the cooling load in humid and tropical climates. Also, investigation of the effects of flow direction in dry and wet channels on performance showed that the performance countercurrent layout is more than other types and using it increases produced cooling load and decreases electricity consumption.

Nomenclature

A	Surface area, m^2
a	Width of wall, m
b	Plate spacing, m
c$_p$	Specific heat at constant pressure, j/kgK
f$_m$	Wettability factor
g	Acceleration due to gravity, m/ s^2
H	Specific enthalpy, j/kg
h	Convective heat transfer coefficient, W/Km^2
h$_m$	Mass transfer coefficient, kg/s m^2
h$_{fg}$	Latent heat of vaporization heat at 0° C, j/kg
k	Thermal conductivity, W/mK
L	Length, m
Le	Lewis number
$\dot{m}$	Mass flow rate, kg/s
Q	Heat flux, W
Re	Reynolds number
RH	Relative humidity
SE	Saturation efficiency
T	Temperature, K
u	Velocity, m/s
U$_z$	Overall heat-transfer coefficient, W/m^{2}K
W	Mass flow rate of water vapor, kg/s

Greek Symbols

δ	Thickness, m
ω	Humidity ratio
μ	Dynamic viscosity, kg/ms
ρ	Dry air density, kg/m^3

Subscripts

db	Dry-bulb
p	Primary air
s	Secondary air
w	Water film
wall	Heat exchanger plate
wb	Wet-bulb
1	Inlet
2	Outlet

References

1. Maclaine, I. L. and Bank, P. L., "A general theory of wet surface heat exchangers and it's application to regenerative cooling", ASME Journal of Heat Transfer, pp. 578-585, (1981).
2. Kettleborough, C. F. and Hsieh, C.S., "The thermal performance of the wet surface plastic plate heat exchanger used as an indirect evaporative cooler", ASME Journal of Heat Transfer, pp. 366–373, (1983).
3. Erens, P. J. and Dreyer, A. A., "Modelling of indirect evaporative coolers", International Journal of Heat and Mass Transfer, Vol. 36, No1, pp. 17–26, (1993).

4. Navon, R. and Arkin, H., "Feasibility of direct-indirect evaporative cooling for residences, Based on Studies with a Desert Cooler", Building and Environment, Vol. 29, No3, pp. 393–399, (1994).

5. Klitsikas, N., Santamouris, M., Argiriou, A. and Asimakopoulos, D. N., "Performance of an Indirect Evaporative Cooler in Athens", Energy and Building, Vol. 21, pp. 55–63, (1994).

6. Stoichkov, N. J. and Dimitrov, G. I., "Effectiveness of cross flow plate heat exchanger for indirect evaporating cooling", International Journal of Refrigeration, Vol. 21, No6, pp. 463–471, (1998).

7. Guo, X. C. and Zhao, T. S., "A parametric study of an indirect evaporative air cooler", Heat and Mass Transfer, Vol. 25, No. 2, pp. 217-226, (1998).

8. Joudi, K. H. and Mehdi, S. M., "Application of indirect evaporative cooling to variable domestic cooling load", Energy Conversion and Management, Vol. 41, pp. 1931–1951, (2000).

9. Ren, C. and Yang, H., "An analytical model for the heat and mass transfer processes in indirect evaporative cooling with parallel/ counter flow configurations", International Journal of Heat and Mass Transfer, Vol. 49, 617–627, (2006).

10. Hettiarachchi, H. D. M., Golubovic, M., Worek, W. M., "The effect of longitudinal heat conduction in cross flow indirect evaporative air coolers"; Applied Thermal Engineering, Vol. 27, pp. 1841-1848, (2007).

11. Martin, R. H., "Characterization of a semi-indirect evaporative cooler"; Applied Thermal Engineering, Vol. 29, pp. 2113-2117, (2009).

12. ISO7730, Moderate Thermal Environments Determination of the PMV and PPD Indices and Specification of the Conditions for Thermal Comfort: International Standardization Organization, (1993).

13. ANSI/ASHRAE55, "Thermal Environmental Conditions for Human Occupancy", American Society of Heating: Refrigerating and Air-conditioning Engineers. Atlanta,GA, (1992).

14. Gilani, N. and Shariaty-Niassar, M., "A comparative survey of indirect evaporative cooler for desirable Indoor air condition", Proceeding of the International Conference on Thermal Engineering, Jordan, (2007).