Heat and mass transfer modeling of two stage indirect/direct evaporative air coolers

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Abstract

Two stage indirect/direct evaporative air cooler can provide summer comfort conditions as an environmentally clean, fresh supply air and energy efficient cooling system for some regions of Iran where direct system alone is not suitable. In this paper a general mathematical model using heat and mass transfer principles for evaporative devices has been described. This model then has been adapted for indirect and direct stages. Effects of various operational and geometrical parameters on cooling performance of this cooling system have been investigated. Using this model, merits of indirect/direct cooling system has been presented for some cities of Iran. Results show that indirect/direct cooling systems can provide comfort conditions for some major cities of Iran, which use only water as cooling fuel and consume much less energy in comparison with conventional vapor compression systems.

Keywords: Two stage evaporative cooler, Indirect/direct air cooler

1. Introduction

Cooling the air by evaporation of water is an environmentally clean and energy efficient method for cooling buildings. Evaporative air-conditioning (EAC) technologies are being used increasingly in residential and commercial applications worldwide. EAC technologies-which rely on water as a coolant rather than on chemical refrigerants-are economical to produce and use and have important environmental benefits. This method of cooling can provide comfort conditions in the areas with dry and hot climate. Evaporative air conditioning provides superior indoor air quality in comparison with vapor compression systems since fresh outdoor air is used. Two principle methods of evaporative cooling are commonly used, the direct and the indirect one.

Direct EAC is the simplest, the oldest, and the most widespread form of air-conditioning. This system typically uses a fan to draw hot outside air into a dwelling through a porous wetted medium. Heat is absorbed by the water as it evaporates from the porous wetting medium, and the air thus leaves the EAC at a lower temperature. In fact this is an adiabatic saturation process in which dry-bulb temperature of the air reduces as it is moistened. The principle underlying direct evaporative cooling is the conversion of sensible heat to latent heat. Some of the sensible heat of the air is transferred to the water and becomes latent heat by evaporating some of the water. The latent heat follows the water vapor and diffuses into the air. The wetted medium could be a porous wetted pad consisting of fibers, cellulose papers or a spray of water. The saturation effectiveness of a direct evaporative cooling system best describes the performance of the system. Saturation effectiveness is defined as the difference between the entering and exit dry-bulb temperatures over the wet bulb depression (difference between ambient dry-bulb and wet-bulb temperatures). An efficient wetted pad can reduce the air temperature by as much as 95% of the wet-bulb depression.

In the indirect method a wet surface heat exchanger is used where a nonadiabatic evaporation takes place. As shown in figure 1 two streams of air are used and there are alternative wet and dry passages which are separated from each other. Primary air which flows in dry passages is cooled sensibly without adding water, while the secondary air which flows in wet passages carries away the heat energy from the primary air. The surface of wet passages is wetted by spray water, so that water film evaporates into the secondary air and decreases the temperature of the wall. Therefore heat is transferred from primary to secondary air without the introduction of moisture into the primary air stream. The air leaving the dry side of the cooler has a lower wet-bulb temperature than the ambient. Consequently it would be advantageous to extract a fraction of this cooled air and pass it through the wet side of the heat exchanger instead of using ambient air. This type of cooler is referred to as a regenerative indirect evaporative cooler. If regenerative evaporative cooler units were placed in series, the cooled air leaving the last unit would approach the ambient dew point temperature, but each additional stage would lower the dry side air mass flow rate resulting lower cooling capacity. If the room outlet temperature has a lower wet bulb than ambient it could be used on the wet passage instead of using a fraction of cooled dry side air or ambient air. This is known as recovery and is suitable for the spaces with low latent loads [1].

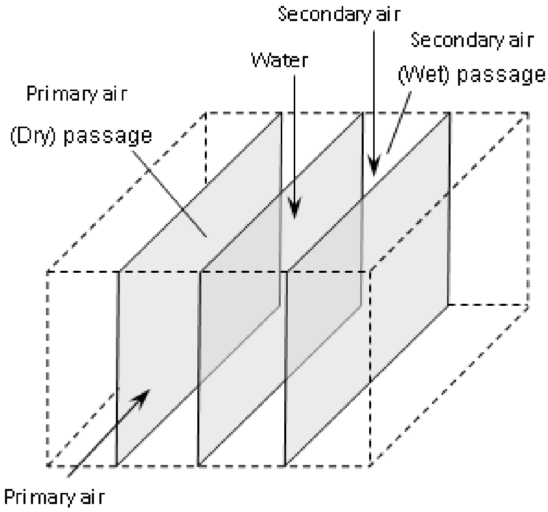

Figure 1. Typical indirect evaporative cooler with cross flow configuration.

Direct evaporative cooling has the disadvantage that if the ambient wet-bulb temperature is higher than 21ºC, the cooling effect is not sufficient for indoor comfort cooling applications. As mentioned above in an indirect evaporative cooler, a heat exchanger is used, meaning that the performance of these types of systems is much lower than that obtained in direct evaporative cooling systems.

Depending on the climatic conditions and the application, combining indirect and direct evaporative coolers might be appropriate to increase the cooling capacity. If a first indirect stage is added to a second direct stage, a two stage indirect/direct cooler is obtained which further cools the air. A two stage air-conditioner combining indirect and direct processes is gaining popularity in places where the higher wet-bulb temperatures (i.e., higher ambient humidity) does not permit sufficiently low indoor temperatures from a simple direct air-conditioner. In this system the outside air is precooled in an indirect stage and then further cooled in a subsequent direct stage. The first stage cools the air without adding moisture and in the second stage moisture is added. The result is that the final air temperature leaving the air-conditioner is generally lower than what could be achieved with a direct air-conditioner only. This expands the application of evaporative air-conditioning considerably to areas with slightly higher wet-bulb temperatures. Commonly 65 percent indirect stage efficiency (performance factor) is reached which allows an ambient wet-bulb temperature of up to 25°C to provide low enough room temperature for real comfort. Two-stage systems are capable of providing cooling that is equivalent to, or even superior to refrigerative air conditioners.

A two stage indirect/direct evaporative cooler is shown schematically in figure 2.This paper deals with the modeling of such a cooler.

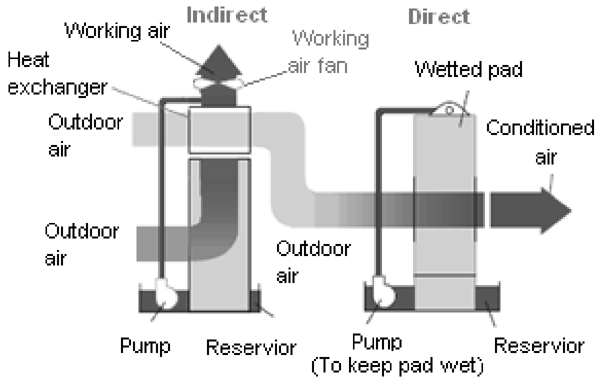

Figure 2. Two stage indirect/direct evaporative cooler

In recent years extensive studies of evaporative cooling systems that include indirect and direct evaporative cooling equipment have been carried out separately and in combination.

As an early work on modeling of indirect cooling, Maclaine-cross and Banks [2] analyzed the evaporative heat transfer in an indirect cooler. They suggested a simplified analysis model assuming that water film is stationary and continuously replenished with water at the same temperature and saturation line is a linear function of temperature. The resulting decoupled equations describing the wet and the dry passages of the cooler were solved by defining a new independent variable: wet bulb depression which is the difference between dry bulb and wet bulb temperatures. This model then could be used to predict the cooler performance by analogy to dry surface heat exchangers.

Hsu *et al.* [3] investigated three basic types of wet surface heat exchangers. They found that cooling effectiveness of each configuration increases with increasing dry channel NTU (Number of Transfer Units) and reaches maximum values asymptotically at some large values of NTU. They assumed that the non-circulating water is locally replenished and its local temperature was calculated from the mass and energy balance equations. They took into account the effect of longitudinal plate conduction. Their results showed that it has almost no effect on the co-current and counter-current configurations and its degrading effect on efficiency of cross flow is accelerated when the ratio of dry-passage length to that of the wet passage is large.

Pescod [4] proposed a simple design method for indirect evaporative cooler using parallel plastic plates with small protrusions. Although the thermal conductivity of plastic is very low, the heat transfer resistance across a thin plastic plate would be less than of the thermal resistance between the air and plate in dry side. Predictions of the efficiencies of Pescod's wet surface plate heat exchanger were found to be higher than the experimental data. Thus incomplete wetting of plate surfaces was suspected.

Kettleborough and Hsieh [5] described a counterflow indirect evaporative cooler with configuration of upward flow of the primary air and downward flows of secondary air and water. Numerical analysis was utilized to study the thermal performance of the unit. By applying wetting factor better agreement between calculated and measured performance data qualitatively was achieved.

Erens and Dreyer [1] reviewed three different models describing evaporative indirect cooler: (1) Poppe model- considering a variable Lewis factor, spray water evaporation rate and modeling over saturation in the secondary air; (2) Merkel model- can be derived from Poppe model by assuming a Lewis factor of unity and negligible effect of spray water evaporation and assuming that the secondary air never becomes over saturated; (3) Simplified model- assuming that the water temperature is constant through out the cooler. They applied these models to a cross flow indirect cooler and simplified model was recommended for small units and for initial design purpose.

Chengqin and Hongxing [6] developed an analytical model for the indirect evaporative cooling with parallel and counter flow configurations. Similar to any other analytical model, humidity ratio of air in equilibrium with water surface was assumed to be a linear function of the surface temperature. Effects of spray water evaporation, spray water temperature variation along the heat exchanger, non unity surface wettability and Lewis factor were considered in the model. Results of analytical solutions were found to be in good agreement with those of numerical integrations.

In the field of direct evaporative cooling, Camargo *et al.* [7] presented the principles of operation for the direct evaporative cooling system for human thermal comfort, and the mathematical development of the equations of thermal exchanges, allowing the determination of the effectiveness of saturation. Dai and Sumathy [8] investigated a cross-flow direct evaporative cooler, in which the wet honeycomb paper constitutes the packing material and the results indicated that there exists an optimum length of the air channel and the performance can be improved by optimizing some operation parameters. Liao and Chiu [9] developed a compact wind tunnel to simulate evaporative cooling pad-fan systems and tested two alternative materials. Al-Sulaiman [10] evaluated the performance of three local natural fibers (palm fiber, jute and luffa) to be used as wetted pads in evaporative cooling.

Most of the previous published researches have been focused on indirect and direct evaporative cooling systems separately.

In comparison with indirect and direct evaporative cooling, there is not much enough reported work on two stage evaporative cooling. El-Dessouky, *et al.* [11] constructed an experimental rig of two-stage evaporative cooling unit and tested in the Kuwait environment. The system was formed of an indirect evaporative cooling unit (IEC) followed by a direct evaporative cooling unit (DEC). During the summer season of Kuwait with dry bulb temperatures higher than 45 $^{\circ}$C the system was operated. The system was operated as a function of the packing thickness and water flow rate of the DEC unit. Other parameters include the water flowrate to the IEC unit and the mode for operating the IEC heat exchangers. Results showed that the effectiveness of the IEC/DEC varies over a range of 90–120%. Similarly, the effectiveness of the IEC unit was varied over a range of 20–40%. The effectiveness of the DEC unit varies over a range of 63–93%.

Scofield and DesChamps [12] studied characteristics of direct and indirect evaporative cooling units, which utilize plate type air-to-air heat exchanger. The first stage of the system contains an indirect evaporative cooling unit, which includes a plate type heat exchanger. In this unit, ambient air, with low wet bulb temperature is sprayed with water before it flows in the plate heat exchanger against indoor air (primary air). This results in reduction of the primary air temperature. Further conditioning of the primary air is achieved in a conventional cooling tower. Operation of this system shows monthly savings of 30% in the energy cost over conventional refrigeration systems.

Al-Juwayhel *et al.* [13] studied the performance of an indirect/direct evaporative cooling system and the effect of coupling the system with a cooling tower. Results show that the highest thermal efficiency is obtained for the combined system, which is followed by a two-stage indirect/direct evaporative cooling unit. The lowest thermal efficiency is reported for the direct evaporative cooling system. In the combined system, the cooling tower removes the thermal load added to the system during air precooling and as a result higher thermal efficiency is achieved.

The above review shows that high efficiency could be obtained by adding an indirect stage prior to direct stage. In this case if abovementioned saturation efficiency is used to evaluate the performance of the two stage unit, values quite higher than those reported in literature for the stand-alone direct evaporative cooling systems, would be obtained. This is because of the cooling effect of the IEC unit. Whereas the efficiencies could be above the unity another definition for cooling effectiveness based on dew point temperature is also used. In this approach, cooling effectiveness is defined as the difference between entering and exit dry bulb temperatures over the difference between ambient dry bulb and dew point temperatures.

2. Modeling

In a two stage evaporative cooling there are two indirect and direct stages. Process in indirect stage is a nonadibatic one with three streams which are primary air (supply air), secondary air (working air) and water. In this stage the balance of mass and energy of all streams should be taken into account. Direct stage undergoes an adiabatic process where only air and water are in contact.

2.1 Physical description of indirect cooler

Indirect evaporative cooler consists of a series of parallel plates in which one is open for primary air flow and the other one is open for water and secondary air flows. Circulation water is sprayed onto the top of the heat exchanger and flows downward along wall surfaces of wet channels. Primary air flows in the alternative channels. As can be seen from figure 1 each repeated section of the cooler consists of a half of dry channel, plate wall and a half of wet channel. For analyzing the processes which occur in an indirect cooler, an element of the surface area dA as shown in figure 3 is considered.

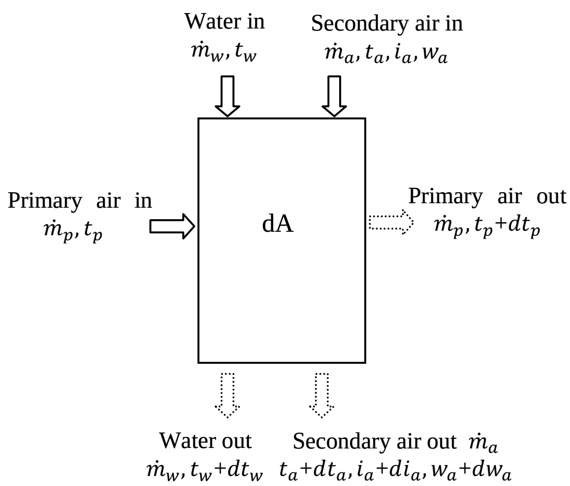

Figure 3. An element of the indirect cooler

The following assumptions here are considered:
-There is no diffusion in the flow direction;
-Cooler is insulated from the surrounding;
-Specific heats are constant;
-Heat and mass transfer coefficients are constant;
-Lewis factor is unity;
-Spray water is circulated;
-Plate wall, bulk water and air/water interface have the same temperature.

2.2 Governing equations of heat and mass transfer

For a differential element as shown in figure 3, by applying principles of mass and energy conservation, a set of differential equations can be obtained as follows.

Energy balance

Energy balance equation for the primary air

$$dq_p = \dot{m}_p c_{p_p} dt_p = -U_0(t_p - t_w)dA \qquad (1)$$

where:
q: Rate of heat transfer (W)
$\dot{m}$: Mass flow rate (kg/ s)
c_p: Specific heat at constant pressure (J/kg K)
t: Temperature (°C)
U_0: Overall heat transfer coefficient (W/m^2 K)
A: Area (m^2)

Subscripts p and w stand for primary air and spray water respectively.

Energy balance equation for the secondary air

$$dq_a = \dot{m}_a di_a = K(i_{asw} - i_a)dA \qquad (2)$$

where:
i: Enthalpy of moist air (J/kg dry air)
K: Mass transfer coefficient (kg/m^2 s)

Subscripts asw and a stand for saturated air at water temperature and secondary air, respectively.

This equation is a result of the assumption of Lewis number ($h_c/(Kc_p)$) being unity. Another approach considers the effect of variable Lewis number. Erens and Dreyer [1] implemented the two methods on a cross flow wet liquid cooler. Their calculation showed insignificant differences between the results by these two methods.

Energy balance equation for the three streams flowing inside the element

With the assumption of constant water flow rate, energy balance equation for the streams flowing inside the element shown in figure 3 is as follows:

$$\dot{m}_p c_{p_p} dt_p + \dot{m}_a di_a + \dot{m}_w c_w dt_w = 0 \qquad (3)$$

Spray water temperature varies inside the heat exchanger. Because of water circulation, the inlet spray water temperature t_{w_1} will equal the outlet spray water temperature t_{w_2}.

$$t_{w_1} = t_{w_2} \qquad (4)$$

Mass balance

The mass balance for the element gives the rate of spray water evaporation $\dot{m}_e$(kg/s)

$$\dot{m}_e = \dot{m}_a dw_a = K(w_{asw} - w_a)dA \qquad (5)$$

where:
w: Humidity ratio of air (kg water/ kg dry air)

Equations (1)-(5) govern heat and mass transfer in an indirect evaporative cooler.

Equations (1)-(3) form a set of coupled ordinary differential equations which can be solved simultaneously, using a multi step numerical integration procedure. Equation (4) sets the condition for inlet and outlet water temperature and this temperature has to be found iteratively. Equation (5) can be solved separately after once solution of the set of equations (1)-(3) has been found, because water temperature is necessary to solve this equation. As in the cross flow configuration, primary air and secondary air flow in perpendicular direction, the heat exchanger is divided into a series of two-dimensional elements. Integration starts from the upmost element in the inlet of primary air and proceeds downward.

Typical modeling has been performed for a cooler of 1m^3 volume and air entering at 25ºC and 50kj/kg enthalpy. Mass flow rates and transport coefficient has been considered from the literature and existing correlations. Figure 4 shows primary air and water temperatures at the inlet and outlet of primary air. As can be seen water temperature at the primary air inlet tends to rise up as flows from top to bottom because here water consecutively encounters a hot column primary air. At the primary air outlet an approximately flat profile of primary air and water temperature could be seen with a difference which is due to the thermal resistance between primary air and water. Averaged profiles of primary air and water temperatures and secondary air enthalpy have been shown in figure 5. In this illustration each variable has been averaged in the direction of its flow. Although water has somehow variations in the heat exchanger, but its averaged value seems to be constant.

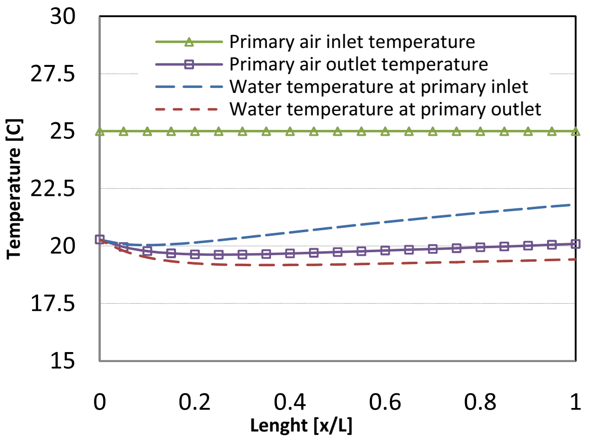

Figure 4. Results of primary air and water temperatures in an indirect cooler.

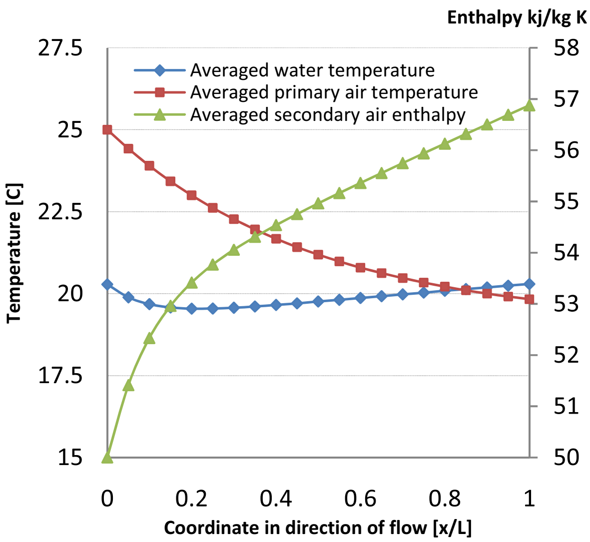

Figure 5. Profiles of averaged temperature

2.2 Physical description of direct cooler

Figure 6 shows a schematic direct evaporative cooling system.

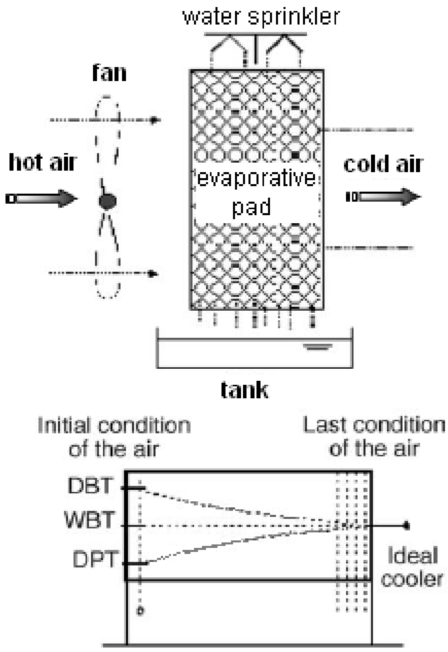

Figure 6. Schematic of a direct evaporative cooler and typical variation of air dry bulb, wet bulb and dew point temperatures through it.

In a direct evaporative cooler, the heat and mass transferred between air and water decreases the air dry bulb temperature (DBT) and increases its humidity, while the enthalpy would be essentially constant in an ideal process. The minimum temperature that can be reached is the wet bulb temperature (WBT) of the incoming air. Wet porous materials or pads provide a large water surface in which the air is moistened and the pad is wetted by dripping water.

Cooling water is sprayed or simply poured from the top to effect falling film at the surface of cellulose papers of pad. Cellulose paper is a good packing for pads because it is porous and durable for repeated wetting and drying, and has a very large pore surface for a given packing volume in comparison to commercial fibers. The main merits of this type of pad are: (1) large pressure loss can be avoided and good effect of evaporation can be ensured, since the process air flows in a straight channel; (2) the evaporative cooler is compact in size and less in weight due to its high surface density.

Considering the humid air flow close to a wet surface, according to figure 7, the heat transfer will occur if the surface temperature t_w is different from the draft air temperature t_a. If the absolute humidity (concentration) of the air close the surface w_{asw} is different from the humidity of the draft w_a, mass transfer will also occur. It is assumed that the makeup water entering the sump to replace evaporated water is at the same adiabatic saturation temperature of the incoming air.

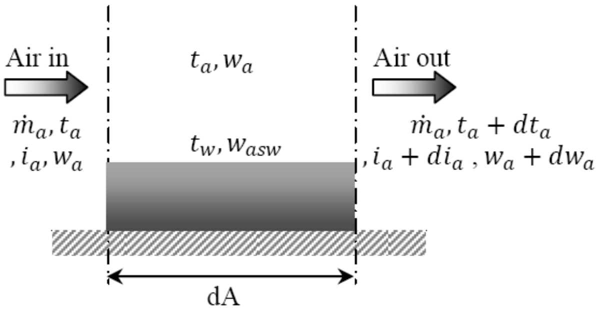

Figure 7. An element of direct evaporative cooler.

For a differential element as shown in figure 7, by applying principles of mass and energy conservation, a set of differential equations can be obtained as follows

Energy balance

Energy balance equation for the air

$$\dot{m}_a di_a = [h_c(t_w - t_a) + K(w_{asw} - w_a)i_{vs}]dA \tag{6}$$

where:

h_c: Convective heat transfer coefficient (W/(m^2 °C))

i_{vs}: Enthalpy of water vapour at water surface temperature (J/kg)

Mass balance

Mass balance equation for the air is the same as equation (5).

In a direct evaporative cooler, water is added in small quantities and attains the wet bulb temperature of the incoming air condition and its temperature is essentially constant within the device. For such a case, taking into account that the water flow direction is irrelevant, and with an additional assumption that Le =1, based on a general mathematical model developed by Halasz [14] the governing equations of mass and energy could be obtained as follows.

$$d(t_a - t_{WB}) = -\frac{h_c}{\dot{m}_a c_{p_a}}(t_a - t_{WB})dA \tag{7}$$

$$d(w_a - w_{WB}) = -\frac{h_c}{\dot{m}_a c_{p_a}}(w_a - w_{WB})dA \tag{8}$$

These equations can be integrated over the whole surface to yield:

$$\frac{t_{a,o} - t_{WB}}{t_{a,i} - t_{WB}} = exp\left(-\frac{h_c A}{\dot{m}_a c_{p_a}}\right) \tag{9}$$

$$\frac{w_{a,o} - w_{WB}}{w_{a,i} - w_{WB}} = exp\left(-\frac{h_c A}{\dot{m}_a c_{p_a}}\right) \tag{10}$$

The effectiveness of a direct evaporative cooling device is defined as:

$$\varepsilon = \frac{t_{a,i} - t_{a,o}}{t_{a,i} - t_{WB}} \tag{11}$$

Then the following equation for effectiveness is obtained.

$$\varepsilon = 1 - exp\,(-\frac{h_c A}{\dot{m}_a c_{p_a}}) \tag{12}$$

This equation shows that an effectiveness of 100% corresponds to air leaving the equipment at the wet bulb temperature of entrance. This requires a combination of large area of heat transfer and a high heat transfer coefficient and low mass flow. Also it is observed that the effectiveness is constant if the mass flow is constant, since it controls directly and indirectly the value of the parameters on the equation.

In analyzing the effectiveness of a direct cooling system, the key point is evaluating the value of $h_c A$. For a rigid cellulose evaporative media, effective evaporative surface area per cubic meter of media could be determined. Therefore according to the size of pad, wetted area in the pad of the equipment would be obtained (Camargo *et al.* [7]).

Dowdy and Karabash [15] presented a correlation to determine the convective heat transfer coefficient in a rigid cellulose paper evaporative media:

$$Nu = 0.1\left(\frac{l_e}{l}\right)^{0.12} Re^{0.8} Pr^{1/3} \tag{13}$$

In above equation l_e is characteristic length and defined by:

$$l_e = \left(\frac{V}{A}\right) \tag{14}$$

where:

A: area of the heat transfer surface; total wetted surface area (m^2)

V: volume occupied by the evaporative pad (m^3)

l: pad thickness (m)

Characteristic length is used to calculate the Reynolds (Re) and Nusselt (Nu) numbers.

Figure 8 illustrates the cooling effectiveness of a direct cooler based on equation (12) as a function of number of transfer units. Results from experimental data show that this simple equation for rigid cellulose paper pads is accurate enough.

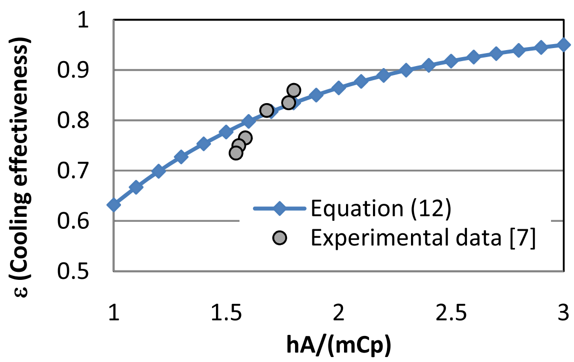

Figure 8. Curve of cooling effectiveness for a direct evaporative cooler versus number of transfer units.

3. Comfort performance of indirect/direct cooling

Iran has a wide variety of climatic conditions. Based on a research [16] the outdoor design conditions for cities of Iran in a period of up to 50 years has been classified according to ASHRAE standard [17]. From these data it has been obtained that for the 1% design conditions, the dry bulb temperature varies from 27 to 49 ºC and wet bulb temperature is in the range of 15 to 33 ºC over the whole country. Such a diverse conditions demand for a variety of cooling systems to meet both the energy saving and comfort conditions requirements. At present in most hot and dry regions direct evaporative cooler is a common cooling system providing comfort for residential buildings. In regions or conditions with high wet-bulb temperature that the direct systems do not provide comfort conditions, vapor compression cooling systems are used. Two stage indirect/direct cooler could provide comfort conditions for a number of cities having a higher wet bulb temperature where direct evaporative cooling is not suitable alone.

Commonly 65% indirect stage cooling effectiveness and 85% direct stage efficiency is attainable. Table 1 illustrates 2.5% design conditions in four major cities of Iran [16].

Table 1: 2.5% design conditions in major cities of Iran.

No.	City	DB[1]	WB[2]	D[3]	I/D[4]
		ºC	ºC	ºC	ºC
(1)	Tehran	37.9	21.8	24.2	19.8
(2)	Isfahan	37.3	20.4	22.9	18.2
(3)	Mashhad	36.1	21	23.3	19.1
(4)	Shiraz	38.2	20.3	23	17.9

[1]Dry bulb temperature, [2]Wet bulb temperature
[3]Direct supply air dry bulb temperature
[4]Indirect/Direct supply air dry bulb temperature

As can be seen from the table two stage indirect/direct cooler could provide comfort conditions for these cities by lowering supply air dry bulb temperature up to 5ºC for this design conditions. It should be noted that for applications which 1% design condition is dictated, the direct evaporative cooler would be even more inapplicable to provide comfort conditions. This brief demonstration shows that indirect/direct evaporative cooler could fill the gap between direct evaporative and vapor compression cooling system as an energy efficient, environmentally clean alternative cooling system providing fresh supply air.

4. Conclusion

Using the general approach to evaporative cooling devices, modeling of a cross flow two stage evaporative cooler has been presented. The indirect stage is well described by the numerical solution of energy and mass balance equations while the direct stage effectiveness is determined by an analytical simple equation which is suitable for cellulose paper pads. Two stage evaporative cooler could provide comfort conditions in some major cities of Iran where direct cooling system is unable to meet comfort requirements.

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