International Journal of Chemical, Materials and Biomolecular Sciences
ISSN: 2415-6620
Vol:7, No:12, 2013

Mathematical Modeling of a Sub-Wet Bulb Temperature Evaporative Cooling Using Porous Ceramic Materials

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Abstract**—**Indirect Evaporative Cooling process has the advantage of supplying cool air at constant moisture content. However, such system can only supply air at temperatures above wet bulb temperature. This paper presents a mathematical model for a Sub-wet bulb temperature indirect evaporative cooling arrangement that can overcome this limitation and supply cool air at temperatures approaching dew point and without increasing its moisture content. In addition, the use of porous ceramics as wet media materials offers the advantage of integration into building elements. Results of the computer show the proposed design is capable of cooling air to temperatures lower than the ambient wet bulb temperature and achieving wet bulb effectiveness of about 1.17.

Keywords**—**Indirect evaporative cooling, porous ceramic, sub-wet bulb temperature.

I. Introduction

THE demand for energy for space cooling in buildings in Middle East, particularly in Gulf cooperation council (GCC) countries has increased tremendously in recent decades. It is estimated that on average about 60% of energy use in buildings is for air conditioning, rising to 80% at peak times. This is mainly driven by cheap and affordable electrical energy, increase in living standards and population growth [1].Globally, the market for air conditioning in buildings is dominated by mechanical vapor compression systems with severe environmental impact related to the increased CO2 emission, the ozone-depleting Chloro Fluro Carbons (CFCs) and localized urban heat islands. In addition, these cooling systems are often linked to health problems affecting work productivity and mental well-being leading to a sharp rise in health complaints such as asthma [2].

Current research efforts are directed towards design of more energy efficiency and environmentally benign alternatives cooling systems in building to reduce energy consumption. One of the natural methods of space cooling in buildings that has been practiced for centuries in hot climates is based on the principle of evaporating water to the surrounding air which known as evaporative cooling. Compared to vapor compression system, evaporative cooling requires lower energy consumption, has lower CO_2 emission and capital and maintenance cost [3].

This paper presents design and mathematical modeling of an indirect evaporative cooling system with a configuration that can be integrated into building elements. The system uses fired clay materials or ceramics that have a highly porous structure to facilitate water evaporation at its surface. The air supply is cooled indirectly to temperatures below prevailing air wet bulb temperature, a system usually known as sub wet bulb temperature indirect evaporative cooling.

II. Evaporative Cooling Process

A common method of direct evaporative cooling for buildings can be found in many parts of Middle East countries including ancient Persian architecture dated from the tenth century and in Egypt in the form of 'Maziara jar' which consists of window screens that were built with holes or niches for water jars. The airflow around the porous jars evaporates the water and depresses its temperature. Evaporative cooling is based on the thermodynamic process of evaporating water to the surrounding air, which involves exchanging sensible heat and latent heat between air and exposed water surface at constant enthalpy. However, changing liquid water to vapor adds moisture to the airflow [4], [5].

Evaporative coolers can be classified into two main categories, direct and indirect evaporative cooling system [6]. Fig. 1 shows a simple diagram of types of evaporative cooling systems.

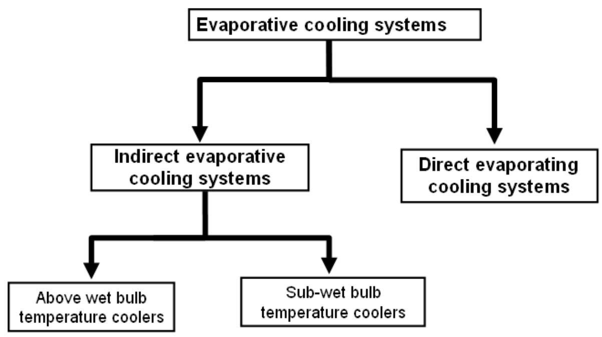

Fig. 1 Types of evaporative cooling systems

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A. Direct Evaporative Cooling

Direct evaporative cooling (DEC) is widely used in hot and dry climates for cooling purposes. In its primitive forms, a wet surface clay pot is used for cooling water and food while a wet mat hanging over a door or window through which hot outside air is blown into the building. Heat from the air is then absorbed by the water as it evaporates from the porous wetting medium, lowering its temperature [7]. The minimum temperature that can be obtained is however limited to the wet bulb temperature of the ambient air. The effectiveness of direct evaporative cooling systems depends largely on a supply air with low relative humidity and reliable source of water to sustain the evaporation process. A typical direct evaporative cooler is shown in Fig. 2 where a fan is used to draw in outside air through a pad wetting media and circulates the cool air through the building [8].

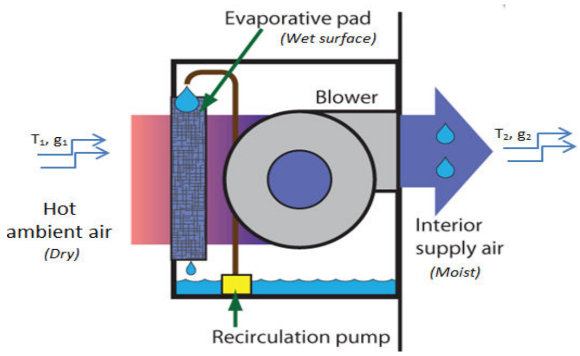

Fig. 2 Schematic of a direct evaporative cooling system [8]

The energy required for evaporation of water is provided by the air, increasing its moisture content and decreasing its temperature. Since the process is adiabatic, the sensible heat loss by the air is balanced out by latent heat gain, which appears as moisture content increase. The heat and mass transfer between the warm dry air and water can be expressed as follows [9]:

$$(\dot{m}_a h_1 + \dot{m}_{v1} h_{v1}) - \dot{m}_{wv} h_{fg} = (\dot{m}_a h_2 + \dot{m}_{v2} h_{v2}) \quad (1)$$

where m_a is the air mass flow rate, h_1 and $h_2$are, respectively, the inlet and the outlet enthalpy. Similarly, m_{v1} is the water vapor inlet mass flow rate, $h_{v1}$is the enthalpy, then, $m_{v2}$is the outlet mass flow rate and $h_{v2}$is the enthalpy.m_{wv} and h_{fg} are the water evaporation rate and latent heat of evaporation respectively.

The amount of water required can be computed as:

$$\dot{m}_{wv} = \dot{m}_{da}(g_2 - g_1) \quad (2)$$

where m_{da} is dry air mass flow rate, g_1 and g_2 are the inlet and outlet air moisture content respectively.

In a well-designed evaporative cooler, the air could be cooled to within 2 to 3°C of the wet bulb temperature [10].

B. Indirect Evaporative Cooling Systems

Indirect evaporative cooling (IEC) reduces the air temperature without increasing its moisture content. This is achieved by cooling the supply air through a heat exchanger which separates it from the wet air stream. Commonly, synthetic thin plate heat exchangers with multiple narrow air channels are used for heat transfer along the air passages [11]. A direct contact between air flow in the wet channel (working air) and water surface increases its humidity, whereas the product air is cooled without humidity change in the dry channel as heat is transferred from the product air to the water film through a water proof thin wall. Fig. 2 shows an example of an IEC where working air is used to evaporate water in a plate heat exchanger which in turn cools the product air. A secondary wet pad is also used to control product air humidity.

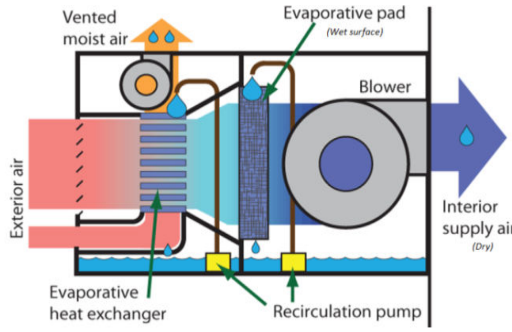

Fig. 3 Schematic of an indirect evaporative cooling system [8]

Similar to DEC, product air in an indirect evaporative cooling system cannot be cooled below its wet bulb temperature. This limitation has prompted researchers to develop and modify the thermal process of direct and indirect evaporative cooling systems to achieve sub-wet bulb temperature evaporative cooling [12].

C. Sub-Wet Bulb Temperature Evaporative Cooling Systems

In a sub-wet bulb indirect evaporative cooling arrangement, the air streams are separated into dry channel for supply air and wet channel for rejecting spent working air. The supply air in the dry channel is cooled indirectly by transferring its heat to the working air in the wet channel through a thin non-permeable channel wall. To achieve sub-wet bulb temperature, part of the cool air in the dry channel is diverted to accomplish the evaporation process in the wet channel, as shown in Fig. 4.

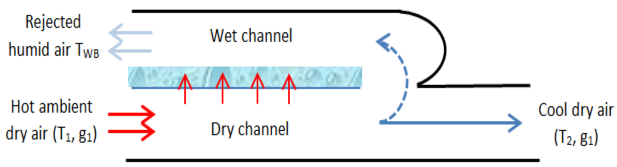

Fig. 4 A simple schematic of a sub wet bulb temperature indirect evaporative cooler

Several mechanical arrangements and thermal performances of sub-wet bulb temperature evaporative cooling systems have been investigated. For instance, Hsu et al. [13] studied the generation of sub-wet bulb temperature cooling by a counter flow and cross flow using two configurations of closed-loop wet surface heat exchangers. They indicated that for the counter flow closed-loop configuration, the maximum wet bulb effectiveness is 1.3 and is reached at a dry passage number of transfer units (NTU) of 10, while for the cross flow closed-loop configuration; the same maximum effectiveness is reached at NTU of 15. Crum et al. [14] indicated that the sub-wet bulb temperature evaporative cooling is achievable by using multistage system with a cooling tower-heat exchanger combination. They proved that this combination has the greatest thermal potential for air conditioning applications since it can produce lowest temperatures and highest cooling capacities for any value of fraction of inlet air delivered. They also concluded that the coefficient of performance for this equipment can reach 75% in the range of air states seen in air conditioning practice. Boxem et al. [15] presented a model for an Indirect Evaporative Cooler which is a compact counter flow heat exchanger with louver fins on both sides. The authors indicated that their calculations overestimated the cooler performance by 20% for inlet air temperatures below 24°C and by 10% for higher inlet temperatures.

Zhao et al. [16], [17] presented a numerical study of a sub-wet bulb temperature counter flow Indirect Evaporative Cooler. A range of design conditions was suggested to maximize the cooler performance: inlet air velocity 0.3-0.5 m/s, height of air passage 6mm or below, length-to-height ratio of air passage 200 and working-to-intake air ratio around 0.4. They concluded that the cooler can give wet bulb effectiveness of up to 1.3 under the UK summer design conditions. Anisimov et al. [18], [19] proposed a combined parallel and regenerative-counter flow Indirect Evaporative Cooler. Based on a mathematical analysis, they indicated that such a cooler would have higher efficiency than other types. Riangvilaikul et al. [20] presented experimental results for a sensible evaporative cooling system at different inlet air conditions (temperature, humidity and velocity) covering dry, temperate and humid climates. The results showed that wet bulb effectiveness ranged between 92 and 114%. A continuous operation of the system during a typical day of summer season in a hot and humid climate showed that wet bulb effectiveness was almost constant at about 102%. Hasan [21] proposed four types of cooler configurations to achieve sub-wet bulb temperature: Two-stage counter flow cooler, Two-stage parallel flow cooler, Single-stage counter flow regenerative cooler and Combined parallel-regenerative cooler, their performance is then compared, a computational model based on mathematical analysis of the heat and mass transfer process inside a cooler is developed. He concluded that with higher number of staged coolers, the ultimate temperature to be reached is the dew point of ambient air.

III. Performance Analysis of a Porous Ceramic Sub-Wet Bulb Temperature Evaporative Cooler

A. Description of the Proposed System

In this work, porous ceramic panels were selected to be used as wet media material in the sub wet bulb temperature evaporative cooler to be integrated into building elements. Ceramic materials are structurally stable and readily availability in many shapes and designs that lend themselves well for cooling applications in buildings. Ceramic materials in direct evaporative coolers have previously been investigated by Ford et al. [22] and Riffat [23].

In the present design, porous ceramic materials in the form of hollow flat shells were filled with water and used as wet media in a sub-wet bulb temperature evaporative cooler, as shown in Fig. 4.

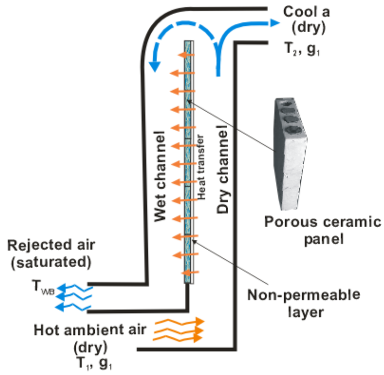

Fig. 5 A schematic of the sub-wet bulb temperature porous ceramic evaporative cooler

The porous ceramic panels were placed between the dry and wet air ducts to form small and narrow ducts with air flowing at low velocity. The dry channel side of the porous ceramic panel is sealed with a thin non-permeable membrane while the wet channel side allows water to sip through its micro-pores onto its surface forming a thin water film. This allows direct contact with the airflow and hence causing water evaporation.

The air streams in the dry and wet channel flow in counter flow arrangement and the supply air exchanges sensible heat with the water in the porous ceramic panels that in turn are cooled through water evaporation on the wet channel side. This results in a drop in temperature of the air in the dry channel without changing its moisture content while the air in the wet channel is rejected at saturation state.

B. Mathematical Modeling

A one-dimensional model was developed to calculate the local distributions of temperature, enthalpy and humidity inside the evaporative air cooler. The air ducts length (L) were divided into small elements (finite volumes) in which the heat and mass transfer equations were applied along the flow direction as shown in Fig. 6. To each element of length (dx), containing three nodes (on dry air, wet air and water film), heat and mass balance equations were applied.

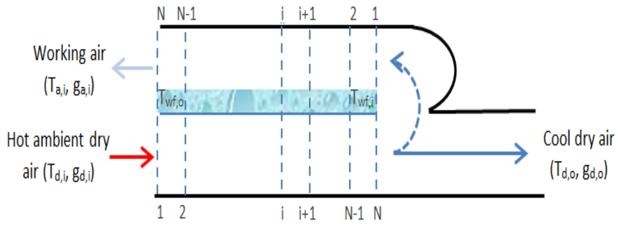

Fig. 6 Computer model set up using finite control volumes

In developing the mathematical model, it was assumed that the cooler is well insulated from its surroundings; thermal conduction in the wall and water film in the x-direction is neglected; the heat and mass transfer coefficients inside each air passage remain constant.

Simultaneous heat and mass transfer processes are described by a system of non-dimensional differential equations; giving the steady state properties of the air in each channel.

- Energy Conservation in the Dry Channel

The Heat transfer from the air stream inside the dry channel to the Water film is given by:

$$\frac{m_d C_{pa}}{D}\frac{\partial T_d}{\partial u} = -K(T_d - T_w) \quad (3)$$

where md is the air flow mass rate in the dry channel, C_{pa} is the specific heat capacity of air, K is the heat transfer coefficient between dry air and the water film, D is the channel depth, T_d is the dry air temperature, T_w is the water film temperature and, u, is the coordinate along the air flow channel length.

- Energy Conservation in Wet Channel

The heat transferred mechanism in the wet channel is more complicated than in the dry channels, as sensible and latent heat is exchanged between the air flow and the water film on the surface of the porous ceramics. This expressed as:

$$\frac{m_a}{D}\frac{\partial h_a}{\partial u} = -\alpha(T_w - T_a) + \beta(g_w - g_a)h_{vw} \quad (4)$$

where α is the convective heat transfer coefficient between the water film and the of air stream in wet channel. This is determined using Nusselt number relationship as:

$$\alpha = Nu\frac{k}{d} \quad (5)$$

Nu is the Nusselt number, k is thermal conductivity of air and, d, is the hydraulic diameter of the channel.

Similarly, the mass transfer coefficient, β in (4), is determined using Lewis number correlation as:

$$Le = \frac{\alpha}{\beta c_p} \quad (6)$$

where Lewis number, Le, value ranges from to 0.9 to 1.15 [22] and to simplify the analysis it is often taken to be 1, c_p is specific heat of humid air and is given by:

$$C_p = C_a + g.C_v \quad (7)$$

- Mass Conservation in Wet Channel

The water evaporation from the ceramic surface appears as an increase in the wet channel air relative humidity (Fig. 9). The mass balance for the water vapor inside the wet channel is then given as:

$$\frac{m_a}{D}\frac{\partial g_a}{\partial u} = \beta(g_w - g_a) \quad (8)$$

- Overall Energy Balance

The overall energy balance between the dry channel airflow, the wet channel air flow and the water film can be expressed as:

$$\frac{m_w c_{pw}}{D}\frac{\partial T_w}{\partial u} = U(T_a - T_w) - \beta(g_w - g_a)(h_{vw} - h_{lw}) - \alpha(T_w - T_a) \quad (9)$$

Furthermore, it was assumed that the moisture content of air in equilibrium with water surface is assumed to be a linear function of the water surface temperature, which is expressed as follows [24]:

$$g_w = a + bT_w \text{ and } g_{wb} = a + bT_{wb}$$

C. Boundary Conditions

The computer model also includes a set of boundary conditions to be satisfied by the solutions of the differential equations. In this case, the air inlet temperature and relative humidity are known parameters and set to ambient prevailing conditions whereas the air inlet to wet channel has the same properties of the air leaving the dry channel. To eliminate any transient effect, the temperature of added water to make up for evaporation loss is supplied at a temperature close to that of the of the rejected air outlet [25].

D. Flowchart of the Computer Model

The sub-wet bulb temperature evaporative cooler was modeled using common energy and mass conservation equations. A flowchart showing the algorithm of different steps of the model program is presented below:

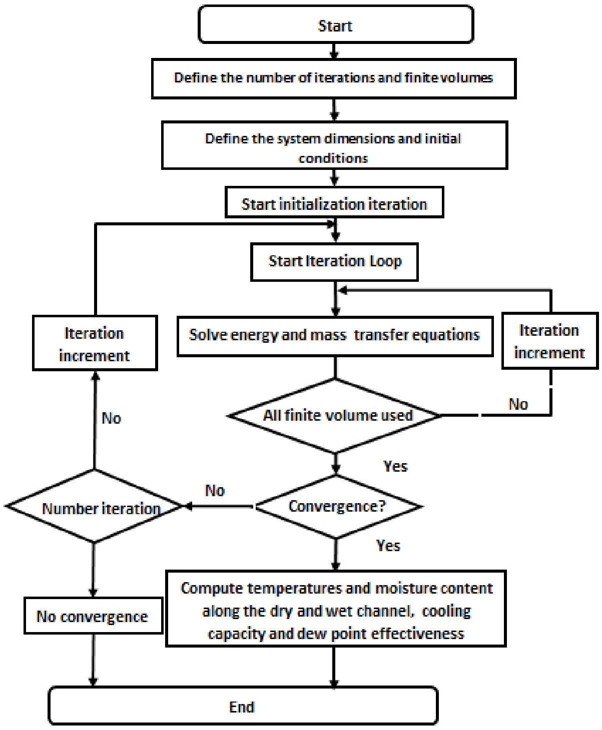

Fig. 7 Heat and mass transfer algorithm flowchart

IV. Results and Discussion

The computer modeling was performed using MATLAB software. The discretized differential equations were applied to each finite volume element with initial condition criteria.

Calculation of the air flow operating parameters along the ducts length was performed until satisfactory convergence conditions. The results of the converged solution includes temperature profiles of air along the dry and wet channel, air moisture content along the wet channel and temperature profile of the water film on the ceramic panel surface.

The model was defined by the parameters given in Table I.

TABLE I
Design and Modeling Parameters

Length	0.5 m
Width	0.5 m
Height	0.35 m
Mass flow rate (dry channel)	0.0014 (kg/s)
Mass flow rate (wet channel)	0.0009 (kg/s)
Air flow regime	Laminar
Nusslet Number	4.861

The computer model results of product air temperature distribution, T_{do}, working air, T_{ao}, and water film, T_{wo}, in the sub-wet bulb temperature evaporative cooling systems are shown in Fig. 8.

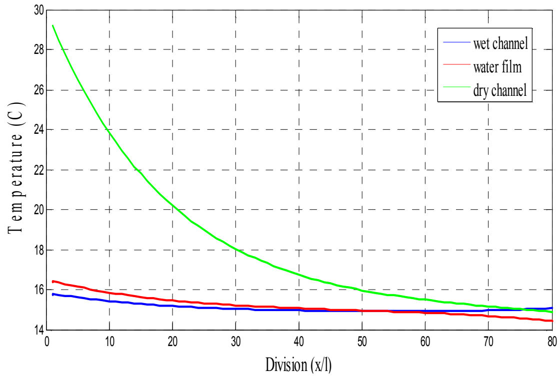

Fig. 8 Air temperature variation along the dry and wet channel

The evolution of ducts temperature was calculated for an initial inlet air temperature of 30°C and relative humidity of 35% (i.e., wet bulb temperature of 18.8°C and dew point of 12.5°C).

It can be seen that the temperature of the air in the dry channel decreases from 30°C to 14.9°C. It reaches a value below the wet bulb temperature value ($T_{wb,i}$=18.8°C), showing that the evaporative cooler would perform adequately in such climatic conditions.

The water film temperature increases from 14.4°C to 16.5°C along the duct as the balance between heat gain from the air in the dry channel and heat loss by evaporation to the air in the wet channel is positive. The heat loss through evaporation of water to the air in the wet channel is due to the increase in the airflow temperature and water content profiles along the channel.

The working air temperature at the turning point of the airflow is higher than that of the water film ($T_{a,i}$=14.94°C >$T_{w,i}$=14.4 °C) , they intersect at Ta=Tw= 14.7°C and then the working air temperature drops to 14.6°C which is below that of the water film before resuming the normal trend and then rejected at a temperature of 15.4°C.

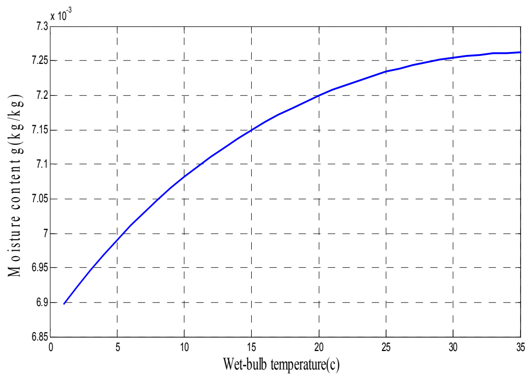

Fig. 9 The evolution of the moisture content in the wet channel

In the wet channel, the evaporative fluid takes heat and stores it in the working fluid as increased latent heat which appears as moisture content increase from 0.0069 kg/kg to 0.00726 kg/kg as shown in Fig. 9.

The cooling capacity of the evaporative cooler per unit area of the available wet surface can be calculated from the following:

$$q = m_s C_{pd}(T_{di} - T_{do}) \quad (10)$$

where q is the specific cooling capacity (W/m^2), m_s presents the supply air mass flow rate (m_d-m_a), C_{pd} is the specific heat capacity of the supply air, T_{di} and T_{do} are the inlet and outlet temperature of the dry channel air flow.

The lowest supply air temperature T_{do} that can be achieved depends on the design of the evaporative cooler (wet area available, porous materials properties, air mass flow rates, etc.) and prevailing climatic conditions particularly relative humidity.

The effect of varying inlet air relative humidity on the cooling capacity is further modeled. Four inlet air temperatures are considered: 30^{o}C, 35^{o}C, 40^{o}C and 45^{o}Cwith relative humidity set at 30%, 40%, and 50%. With supply air flow rate remaining constant at 1.5 kg/h. Fig. 10 shows that the cooling capacity is inversely proportional to the relative humidity. However, it increases with increasing inlet air temperature. At high inlet temperatures and low relative humidity, cooling capacity can be as high as 120 W/m^2.

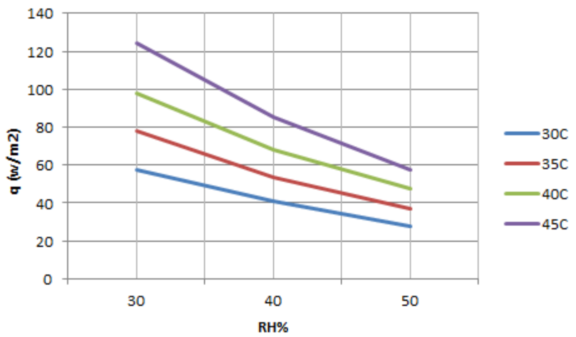

Fig. 10 Cooling capacity for different Temperature and Relative Humidity

Finally, the effectiveness of the evaporative cooler was also evaluated using two different methods: the wet bulb and dew point effectiveness.

The wet bulb effectiveness is the ratio of the difference between inlet and outlet air temperature to the difference between inlet air temperature and its wet bulb temperature [26]. The mathematic expression of the wet bulb effectiveness is given by:

$$\varepsilon_{wb} = \frac{T_{db,i} - T_{db,o}}{T_{db,i} - T_{wb,i}} \quad (11)$$

The dew point effectiveness, on the other hand, is the ratio of the difference between inlet air temperature to the difference between inlet air temperature and its dew point temperature. The mathematic expression of the dew point effectiveness is expressed as:

$$\varepsilon_{db} = \frac{T_{db,i} - T_{db,o}}{T_{db,i} - T_{dp}} \quad (12)$$

For the design inlet air conditions of 30^{o}C and 35% relative humidity the web bulb effectiveness is ε_{wb}=1.23 and dew point effectiveness is ε_{db}= 0.779 . This shows that the sub wet bulb temperature evaporator cooler has wet bulb effectiveness higher than unity, a thermal performance that can compare favorably with more mechanical vapor compression systems and can contribute to reducing overall energy consumption for air condition in buildings.

These results shows that an ideally designed system would achieve adequate supply temperature for space air conditioning in hot and dry climatic conditions, such as that prevailing in large part of the Middle East countries. The adoption of ceramic materials, as wet media, would have additional benefit in reliability and compatibility for integration in building elements.

V. Conclusions

Modeling of an indirect evaporative cooling system for supply of sub wet bulb air temperature was carried out. It was shown that an ideally designed system would achieve adequate supply temperature for space air conditioning in hot and dry climatic conditions. It was concluded that the proposed method for indirect evaporative cooling is capable of cooling air to temperatures lower than the ambient wet bulb temperature. The wet bulb cooling effectiveness ε_{wb} reached 1.23. It was shown the cooling capacity is strongly dependent on the ambient air temperature and relative humidity. The maximum cooling capacity achieved at a temperature of 45^{o}C and relative humidity of 30% was about 120 W/m^2 of wet ceramic material wet area.

Acknowledgment

This publication was made possible by NPRP grant No. 4 - 407 -2 -153 from the Qatar National Research Fund (a member of Qatar Foundation). The statements made herein are solely the responsibility of the author.

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Dr. Rabah Boukhanouf has over 15 year experience in teaching and research in the area of energy efficient and sustainable technologies including combined heat and power, heat pumps, passive heating and cooling technologies and ventilation in buildings. He has worked with industry on many projects and is named as the inventor on 6 international patents. He has also published over 40 papers in referred journals and international conferences and supervises several PhD students and research assistants.

Dr. Hatem G. Ibrahim holds the position of Associate Professor at Qatar University. Dr. Ibrahim has a long and established research experience including managing green construction, carbon abatement in construction industry using knowledge based programming, and preservation of traditional architectural and urban heritage of Qatar. The latter being particular an ass-on advantage for reconciling the integration of new low carbon technologies with the traditional architectural concepts.