INVESTIGATION OF A SUB-WET BULB TEMPERATURE EVAPORATIVE COOLER FOR BUILDINGS

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Abstract. The paper presents a computer model and experimental results of an indirect evaporative cooling system for air conditioning in hot and dry climate regions. The system uses porous/fired clay materials as wet media for water evaporation. The supply air and working air flows were staged in separate ducts and in counter flow direction. Modelling results were conducted for ambient air dry bulb temperature ranging from 30° C to 45° C and relative humidity lower than 65%, showing that supply air would be cooled to below wet bulb temperature. In addition, it was estimated that the indirect evaporative cooler would achieve cooling capacity of approximately 27 W/m² of exposed ceramic material wet area and with overall wet bulb effectiveness greater than unity. This performance would make the system a potential alternative to conventional mechanical air conditioning systems in buildings.

1 INTRODUCTION

Energy consumption in buildings is stands at between 30–40% of the total primary energy use globally (Dodoo et al., 2011). A major part of this is used to provide comfortable indoor climatic conditions for occupants. For example, in regions with cold climates such as northern Europe, energy for space heating and hot water accounts for over 60% of the total energy used in buildings, whereas in hot climate region a similar proportion of energy consumption is for air cooling in buildings (Proctor, 2010). The growth in air conditioning systems in the world is mainly driven by an increase in living standards, affordability, population increase and cheap electrical energy in some regions such Middle East. This has led many countries to build new power generation plants and extend grid infrastructure to meet peak electricity loads, which in turn impacts negatively on the environment by emitting more greenhouse gases.

Current air conditioning market is dominated by mechanical vapour compression systems, which are energy intensive systems and suffer from low thermal performance in hot climate conditions. Hence, in hot dry climate region, application of low carbon cooling technologies is once more the focus of many researchers including methods of integration into modern buildings and materials that are more durable.
2 EVAPORATIVE COOLING TECHNOLOGY

The use of evaporative cooling for thermal comfort in buildings is not new. The earliest use of evaporative cooling was by ancient Egypt and the Roman Empire using for example wet mats (cooling pads) over doors and windows to cool the indoor air when wind blew through the mats (Vissers, 2011). Evaporative cooling is widely found in Middle East and Persian architecture often integrated into the windcatchers. Evaporative cooling is a low carbon and economically feasible method for cooling buildings in hot and dry climates.

2.1 Direct evaporative cooling systems

Direct evaporative cooling is the process of evaporating liquid water to the surrounding air and causing its temperature to decrease. A typical direct evaporative cooler, as shown in Figure 1, uses a fan to draw in outside air through a pad wetting media and circulates the cool air through the building.

The energy required for evaporation of water is provided by the air, though at the expense of 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 (Jones, 2001).

\[
(m_a h_1 + \dot{m}_v h_{v1}) - \dot{m}_w h_{gw} = (m_a h_2 + \dot{m}_v h_{v2})
\]

where \(m_a\), \(h_1\) and \(h_2\) are air mass flow rate, inlet and outlet enthalpy respectively. \(m_v\), \(h_{v1}\), \(m_v\) and \(h_{v2}\) are the water vapour inlet mass flow rate, enthalpy, outlet mass flow rate and enthalpy respectively. \(m_{gw}\) and \(h_{gw}\) are the water evaporation rate and latent heat of evaporation respectively. The amount of water required can be computed as:

\[
\dot{m}_{gw} = \dot{m}_{da} (g_2 - g_1)
\]

where \(\dot{m}_{da}\) is dry air mass flow rate, \(g_1\) and \(g_2\) are the inlet and outlet air moisture content respectively.

The effectiveness of direct evaporative coolers is primarily influenced by the air wet bulb temperature and in a well-designed system the air could be cooled to within 2 to 3°C of the wet bulb temperature which presents a severe thermodynamic limitation.
2.2 Indirect Evaporative cooling and sub wet bulb temperature

This has led several researchers to develop and modify the thermal process of direct evaporative cooling system to achieving sub-wet bulb temperature, referred to as Dew point or Sub-wet bulb temperature evaporative cooling (Maisotsenko, 2003). In this coolers 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 Figure 2.

![Figure 2: A simple schematic of a sub wet bulb temperature indirect evaporative cooler](image)

The advantage of this arrangement is that the moisture content of the cooled air remains unchanged. Hsu et al. (1989) carried out a theoretical and experimental study on two configurations of closed-loop wet surface heat exchangers to achieve sub-wet bulb temperature cooling through counter flow and cross flow air stream arrangements. Boxem et al. (2007) presented a model for a compact counter flow Indirect Evaporative Cooler with finned exchanger. The performance of a 400 m$^3$/h air flow rate cooler was analysed and showed that for inlet air temperatures higher than 24 C the model results accuracy were within 10%. Zhao et al. (2008) presented a numerical study of a counter flow Indirect Evaporative Cooler for sub-wet bulb temperature cooling. The authors suggested a range of design conditions to maximize the cooler performance including air velocity range, height of air passage, and length to height ratio of air flow rates. It was shown that even under UK summer conditions the cooler can yield wet bulb effectiveness of up to 1.3. Riangvilaikul et al. (2010) 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 show 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 (2010) also presented a theoretical model of four different configurations of indirect and sub wet bulb temperature coolers: two-stage counter flow cooler, two-stage parallel flow cooler, single-stage counter flow regenerative cooler and combined parallel-regenerative cooler. The author concluded that with higher number of staged coolers, the ultimate temperature to be reached is the dew point of ambient air.

2.3 Wet media materials

The wet media used in evaporative coolers is an essential component of an evaporative cooler. It is usually made of a porous material with large surface area and capacity to hold
liquid water. According to Wanphen and Nagano (2009), the selection of wet media materials is based on their effectiveness, availability, cost, safety, and environment factors. Zhao et al. (2009) investigated various types of porous materials such as metal and plastic foams, zeolite and carbon fibres to be used as wet media for heat and mass transfer in evaporative cooling systems. Musa (2008) also investigated the use of more common aspen pads materials for indirect evaporative cooling system. Riffat and Zhu (2004) employed ceramic materials for indirect evaporative cooling systems. Figure 3 shows some common wet media materials that can be found in evaporative cooling systems.

![Figure 3: Wet media materials](image)

3 DESIGN OF A POROUS CERAMIC SUB-WET BULB TEMPERATURE EVAPORATIVE COOLER

3.1 Description

In this project, porous ceramic materials in the form of hollow flat shells were used as wet media in a sub-wet bulb temperature evaporative cooler. Porous ceramic materials were selected for their stable structural, non-corrosion properties and easily moulded into desired shape. Figure 4 shows the configuration of the sub-wet bulb temperature evaporative cooler using the porous ceramic material for water evaporation.

![Figure 4: A schematic of the sub-wet bulb temperature porous ceramic evaporative cooler](image)
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.

3.2 Mathematical model

The sub-wet bulb temperature evaporative cooler was modelled using common energy and mass conservation laws. In the model the dry and wet channel were divided into small elements (finite volumes) to which the energy and mass transfer equations were applied.

**Energy conservation in the dry channel**

Air is cooled in the dry channel by transferring its sensible heat to the wet channel through the non-permeable layer and the porous ceramic panels. This can be expressed as follows:

$$\frac{\dot{m}_d \partial h_d}{\partial A} = -U(T_d - T_{fw})$$

where \(m_d\), \(h_d\), \(T_d\), \(A\) and \(U\) are the air flow mass rate, enthalpy, temperature, heat transfer coefficient and area of the dry channel. \(T_{fw}\) is the temperature of the water film on the wet channel side.

**Energy conservation in the wet channel**

The heat transfer mechanism in the wet channel is more complicated than in the dry channels, as sensible and latent heat is exchanged between the airflow and the water film on the surface of the porous ceramics. This is expressed as (Halasz, 1998):

$$\frac{\dot{m}_w \partial h_w}{\partial A} = \kappa(T_{fw} - T_w) + \sigma(g_{fw} - g_w)h_{fg}$$

where \(m_w\), \(h_w\), \(T_w\), \(g_w\), and \(\kappa\) are the air mass flow rate, enthalpy, temperature, moisture content and convective heat transfer coefficient in the wet channel. \(T_{fw}\), \(g_{fw}\), and \(\kappa\) are the water film temperature, saturated air moisture content, and mass transfer coefficient. It is assumed that air flow regime in both dry and wet channel are laminar and the mass transfer coefficient, \(\sigma\) obeys the following Lewis number correlation (Hasan 2010, Halasz, 1998):

$$Le = \frac{\kappa}{\sigma c_p}$$

where Lewis number, \(Le\), value ranges from to 0.9 to 1.15 and to simplify the analysis it is often taken to be 1, \(c_p\) is specific heat of humid air.

**Mass conservation in the wet channel**

Water evaporation from the ceramic panel surface appears as an increase of the air moisture content along the length of the wet channel. The mass balance for the water vapour in the wet channel can be written as:
\[
\frac{\dot{m}_w}{\dot{A}} \frac{\partial g_w}{\partial \sigma} = \sigma (g_{fu} - g_w) \tag{6}
\]

**Overall energy balance**

The overall energy and mass balance at the water film interface between the air flow in the dry channel, the water film on the ceramic surface and the air flow in the wet channel can be expressed as:

\[
\frac{\dot{m}_{fu} C_{pfw} \partial T_{fu}}{\partial A} = U(T_d - T_{fu}) - \sigma (g_{fu} - g_w) h_{fg} - \alpha(T_{fu} - T_w) \tag{7}
\]

where \( C_{pfw} \) is specific heat of water.

The computer modelling was performed using Matlab\textsuperscript{©} software. The governing differential equations were discretised and applied to each finite volume element along the dry and wet channel length. It was assumed that the air properties, heat and mass transfer coefficients are constant in each finite control volume, the water film and the non-permeable membrane thermal resistances are assumed to be negligible.

The initial conditions used in this model include known air properties (temperature and moisture content) for the dry channel and air moisture content for the wet channel. The main design parameters of the system are given in Table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air channel Length</td>
<td>0.5 m</td>
</tr>
<tr>
<td>Air channel Width</td>
<td>0.5 m</td>
</tr>
<tr>
<td>Air channel Height</td>
<td>0.005 m</td>
</tr>
<tr>
<td>Mass low rate in the dry channel</td>
<td>0.0266 (kg/s)</td>
</tr>
<tr>
<td>Mass flow rate in the wet channel</td>
<td>0.01 (kg/s)</td>
</tr>
<tr>
<td>Air flow regime</td>
<td>Laminar</td>
</tr>
</tbody>
</table>

Table 1: design and modelling parameters

Computation of the operating parameters of airflow along the air ducts length was performed iteratively until converging conditions were satisfied giving a temperature difference between two consecutive iterations less than 0.01°C.

**4 Experimental test rig**

A laboratory test rig was built to test the porous ceramic sub-wet bulb temperature evaporative cooler is shown in Figure 5. The porous ceramic panels were filled with water through and overhead tank while a fan was used to draw air at controlled temperature and relative humidity from an environmental chamber and circulate it through the evaporative cooler dry and wet channels. The rig was fully instrumented to measure the air temperature, moisture content and flow rates along the dry and wet channels.
5 RESULTS AND DISCUSSION

The thermal performance of the porous ceramic evaporative cooler design was evaluated using computer modelling and laboratory testing. Figure 6 shows the computer and experimental results of the air temperature profile in the dry and wet channel. For initial inlet air (atmospheric air) dry bulb temperature, $T_{dbi}$, of 30°C and relative humidity of 35% (i.e., wet bulb temperature, $T_{wbi}$, of 18.8°C and dew point of 12.5°C), the computer model predicts that the dry bulb temperature of the air in the dry channel would be cooled to 16.9°C which is 1.9°C below the web bulb temperature. The airflow along the wet channel, on the other hand, increased from 16.9°C to 19.8°C, as the energy balance between sensible heat loss and latent heat gain is positive. The experimental measurements however show the air temperature at the outlet of the dry channel is about 22.8°C and that of the wet channel is 22°C. The discrepancy between the experimental and computer model results could be explained by the difficulties in obtaining a uniform air distribution in the dry and wet channel.
The representation of the state of the air in the dry and wet channels on a psychrometric chart is shown in Figure 7. It can be seen that the air temperature at the dry channel outlet, $T_{do}$, is lower than air wet bulb temperature, $T_{wb}$, at the inlet with the ultimate air supply temperature equals to dew point temperature, $T_{dp}$.

Further evaluation of the performance of the evaporative cooler was carried out by calculating its cooling potential at various dry bulb and relative humidity conditions as shown in Figure 8. It can be seen that the cooling capacity is strongly influenced by the dry bulb temperature and relative humidity of air. The cooling capacity at 40°C and 35% relative humidity is around 27W/m² of the wet porous ceramic area.

Finally, the effectiveness of the evaporative cooler was also evaluated, which can be determined 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 (Riangvilaikul and Kumar, 2010). The mathematic expression of the wet bulb effectiveness is given by:

$$\varepsilon_{wb} = \frac{T_{db, in} - T_{db, out}}{T_{db, in} - T_{wb, in}}$$

(7)
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 (Frank, 2011). The mathematic expression of the dew point effectiveness is expressed as:

$$\varepsilon_{db} = \frac{T_{db,in} - T_{db,out}}{T_{db,in} - T_{dp}}$$

(8)

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

5 CONCLUSION

A computer model and experimental results of a sub-wet bulb temperature evaporative cooler using porous ceramic materials were presented. It was shown that the evaporative cooler can achieve high thermal performance in terms of low air supply temperatures and effectiveness. The structural stability and manufacturing controllability of ceramic materials lend them well to integration into buildings and performing the function of air conditioning in regions with hot and dry climatic conditions.

ACKNOWLEDGEMENT

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 authors.

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