Thermal Model Of A Residential Building With Regenerative Evaporative Cooling System

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Abstract: - Modern society has an increasing dependence on vapor air conditioning and refrigeration systems which consume large amounts of electrical power that is often generated from fossil fueled power stations, with the effect releasing large quantities of greenhouses gases, such as CO2 into the atmosphere. These have led to increased focus on the development of innovative and ‘environmentally-friendly’ air conditioning systems taking advantage from traditional cooling methods. This paper seeks to present regenerative sub wet bulb evaporative cooling methods and to evaluate its performances with residential building thermal model. Heat and mass transfer model is constructed and typical condition are identified and used for the thermal model to evaluate the cooling cost of the system. The findings of this study are relevant to the neo-traditional cooled constructions in hot and dry countries.

Key-Words: Regenerative evaporative cooling, Sub-wet bulb temperature, Thermal model, cooling cost.

1 Introduction

Until recent decades, the only energy available in most societies was what they could find, mine, collect and carry home, be it dung, coal, wood, peat, water or ice. These sources, passively or actively provided for cooling or heating the occupied spaces were developed and added to the survival challenges that the increasingly extreme climate posted.

Looking back over the traditional Middle East buildings, various natural cooling systems are seen in the traditional architecture. Commonly, the architects relies on natural energies, to render the inside condition of the buildings pleasant, such as the use of arched towers, wind catchers, subterranean houses. These low cost and energy efficient technologies provided a successful adaptation to the harsh climate while respecting the environment and the human comfort [1].

To provide comfortable, low carbon and low energy buildings, it is important to consider a whole system as a dynamic three interactive core: climate, people and buildings, as presented in Fig. 1.

Fig. 1 Energy needs traditional three-way interaction (source: Nicol et al, 2012 [3])
This interaction, between the system components remain the challenge of the neo-traditional and modern cooling technologies.

Sensitive solution were developed over a long period of time to the particular characteristics of the environment in which they were constructed. A number of building projects were implemented by using general passive design strategies, traditional architecture elements and original efficient techniques.

Passive Evaporative Cooling is one of the neo-traditional cooling method that uses the evaporation of water to cool the surrounding air. Its application is based on the availability of water resources and use of draughts into the building.

A common method of Evaporative Cooling has been documented in Iranian palaces dating from the tenth century and still can be found in Cairo. This strategy consists of window screens that were built with holes or niches for ‘water jars’. The airflow through these porous jars evaporated the water and [3][5].

To test the performance of the system, an experiment was set by Cain et al. [6]. Water samples was taken at various stages to be tested for purity.

As presented in Fig. 2, the results of the climatic tests showed that for an ambient air temperature ranged from 19 °C to 36°C, over the day, the temperature of the water jar remained relatively constant at 20 °C.

2 Regenerative Evaporative Cooling system: Mathematical model

The proposed passive cooler is an Indirect sub-wet bulb temperature Evaporative Cooling system. Generally, Indirect Evaporative Cooling (IEC) systems are aims to reduce air temperature, of an occupied space, without adding moisture. Thermodynamically, an IEC passes primary air stream over one side of a heat/mass exchanging area, and secondary air over its opposite side. The opposite side, often called ‘wet’ passage, absorbs heat from the ‘dry’ side by evaporating water and, Therefore, cooling the primary air while the latent heat of vaporizing water is released to the wet side air [7]. It is important to underline the effect of the evaporative latent heat, resulting from the vaporization of the liquid film, which plays a major role in the heat transfer process.

To achieve sub-wet bulb temperature, part of the primary product air in the dry passage is diverted to accomplish the evaporation process in the wet passage.

Several mechanical arrangements and thermal performances of sub-wet bulb temperature evaporative cooling systems have been investigated. For instance, Zhao et al. [8] indicated that the sub wet bulb temperature evaporative cooling is achievable by using multistage system with cooling tower-heat exchanger system. Based on the thermal model, they concluded that the proposed cooler has the potential performances for air conditioning applications. Lowest cooling temperatures and highest cooling capacities can be achieved for any value of process air fraction.

Boxem et al. [9] presented a an Indirect Evaporative Cooler with a compact counter flow heat exchanger and louver fins on the sides.
Cooling performances for different inlet air temperature range has been estimated. Numerical model of various counter flow evaporative cooling arrangement has been proposed by Zhao et al. [10][11]. For high cooler performances, optimal working condition has been fixed: 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.

Hasan [12] proposed four stage types of cooler configurations to achieve sub-wet bulb temperature. Their performance has been compared based on a computational model 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.

The proposed IEC is a single stage regenerative sub-wet bulb temperature evaporative cooler. A schematic description is presented in Fig. 3. The counter flow air cooler is mainly composed by: A small fan, two adjacent air passages, separated by very thin non permeable wall, a water film and a small duct to evacuate the rejected air.

The ambient air is pulled, at fixed airflow, inside the dry passage while secondary air stream flows inside the wet passage over the water film. Vapor pressure gradient between the water film and the secondary airstream causes mass transfer, by evaporation, from the saturated water surface to the air. This results in lowering the water film temperature. As a result, temperature gradient is created between the water film and the primary air stream and, therefore, due to the water heat losses, the primary air stream temperature decreases along the dry passage.

The cooled product is supplied to the building without moisture content increase, however, the saturated secondary air is rejected to the ambient. By precooling the secondary airstream, at the wet passage, a sub-wet bulb temperature indirect evaporative cooling process can be achieved.

2.1 Computational Model

One dimensional model was developed to calculate the local distributions of temperature, enthalpy and humidity inside the evaporative air cooler. 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 passage.

Under these assumptions, the energy conservation balance is written as

\[ \frac{m_d C_p d}{A} \frac{\partial T_d}{\partial x} = -K_s (T_d - T_{wf}) \]  
(1)

\[ \frac{m_w \partial h_w}{A} \frac{\partial x}{\partial x} = h_w \beta (g_{wf} - g_w) + \alpha (T_{wf} - T_w) \]  
(2)
The convective heat transfer coefficient between the secondary air stream and the water film is given by the Nusselt Number

$$Nu = \frac{\alpha d_s}{k_a}$$  \hspace{1cm} (3)

Similarly, the mass transfer coefficient, $\beta$ in is given by the Lewis number correlation

$$Le = \frac{\alpha}{\beta c_p}$$  \hspace{1cm} (4)

The conservation of mass equation is expressed as

$$\frac{m_w g_w}{a} \frac{\partial g_w}{\partial x} = \beta (g_{wf} - g_w)$$  \hspace{1cm} (5)

The energy balance at the water film interface is given by

$$\frac{m_w c_p w}{a} \frac{\partial T_w}{\partial x} = U (T_w - T_{wf}) - \beta h_w (g_{wf} - g_w) - \frac{\alpha (T_{wf} - T_w)}{\alpha (T_{df} - T_w)}$$  \hspace{1cm} (6)

### 2.1.1 Nondimensionalization

With the transformation $\tilde{T} = (T - T_{wb}) \tau$ and $\tilde{g} = (g - g_{wb}) \sigma$, the passage temperature and moisture content may be written in terms of nondimensional parameters as [13]

$$\tilde{T}_d = \frac{T_{d,in} - T_{wb}}{T_{d,in} - T_{wb}}, \quad \tilde{T}_w = \frac{T_{w,in} - T_{wb}}{T_{w,in} - T_{wb}}, \quad \tilde{T}_{wf} = \frac{T_{wf} - T_{wb}}{T_{d,in} - T_{wb}}$$  \hspace{1cm} (7)

Similarly, the nondimensional moisture content ratios are

$$\tilde{g}_w = \frac{g_w - g_{wb}}{g_{wb} - g_{d,li}}, \quad \tilde{g}_{wf} = \frac{g_{wf} - g_{wb}}{g_{wb} - g_{d,li}}$$  \hspace{1cm} (8)

### 2.1.2 Numerical Discretization

Considering a finite volume of the single stage regenerative cooler, governing equation in differential form may be written in finite difference discretization form.

Considering nondimensional space coordinate, the above system of equation (1)-(7), can be transformed to:

$$\frac{\partial \tilde{T}_d}{\partial u_d} = -K \chi C (\tilde{T}_d - \tilde{T}_{wf})$$  \hspace{1cm} (9)

$$\frac{\tilde{T}_{d+1,j} - \tilde{T}_{d,j}}{\Delta u} = -KC\chi \frac{\tilde{T}_{d+1,j} - \tilde{T}_{d,i}}{2} + KC\chi \frac{\tilde{T}_{wf+1,j} - \tilde{T}_{wf,i}}{2}$$  \hspace{1cm} (10)

With nondimensional coefficients:

$$K = \frac{K_S}{\alpha}, \quad C = \frac{m_a c_p a}{m_d c_p}, \quad \chi = \frac{a A_s}{m_a c_p}$$  \hspace{1cm} (11)

Considering equation (8), non-dimensional humidity ratio in the wet passage is transformed to

$$\frac{\partial \tilde{g}_w}{\partial u_w} = - \frac{L_e A_s}{m_a c_p} \tilde{g}_w + L_e B \frac{L_e A_s}{m_a c_p} \tilde{T}_{wf}$$  \hspace{1cm} (12)

As detailed below, the surplus variable $B$ is obtained by the linear relationship between the temperature and the moist airstream in the wet passage:

$$\tilde{g}_w = a + b \tilde{T}_w \quad \text{and} \quad \tilde{g}_{wb} = a + b \tilde{T}_{wb}$$  \hspace{1cm} (13)

Where $b$ is obtained as

$$b = \frac{\tilde{g}_w - \tilde{g}_{wb}}{\tilde{T}_w - \tilde{T}_{wb}}$$  \hspace{1cm} (14)

Using Equations (7)-(8), the humidity ratio of the saturated secondary air is given by:

$$\tilde{g}_w = \frac{\beta c_p}{\alpha} \frac{br_{wb}}{c_p} \tilde{T}_w = Le B T_w$$  \hspace{1cm} (15)
Assuming that \( r_{wb} \) and \( c_{pa} \) are almost constant, the magnitude of B depend on b which depends on both \( Ť_w \) and \( T_{wb} \) [13].

Transforming equation (12) into the discretised form:

\[
\frac{\theta_w - \theta_{w_{i,j}}}{\Delta U} = -L_e \chi \frac{\theta_{w_{i+1,j}}}{{2}} - L_e B \chi \frac{T_{w_{i+1,j}} + T_{w_{i,j}}}{2}
\]  

On the other hand, the enthalpy of unsaturated secondary moist air is given by

\[
\frac{\partial h_w}{\partial u_w} = (c_{pw} + g_w c_{pv}) \frac{\partial T_w}{\partial u_w} + (r_{w0} + c_{pw} T_w) \frac{\partial g_w}{\partial u_w}
\]

Substituting \( \frac{\partial g_w}{\partial u_w} \) from equation (12) and \( \frac{\partial h_w}{\partial u_w} \) from equation (2), the nondimensional form of the secondary air stream temperature is easily simplified to:

\[
\frac{\partial T_w}{\partial u_w} = -\chi \left(T_w - T_{wf}\right)
\]  

\[
\frac{T_{w_{i+1,j}} - T_{w_{i,j}}}{\Delta U} = -\chi \frac{T_{w_{i+1,j}} + T_{w_{i,j}}}{2} + \chi \frac{T_{w_{i+1,j}} + T_{w_{i,j}}}{2}
\]

Finally, using the previous transformation, the water film temperature can be simplified as:

\[
\frac{\partial T_{wf}}{\partial u_w} = -\chi \left(\frac{K}{w} \tilde{T}_{d} + \frac{K}{w} \tilde{T}_{w} + \frac{K}{w} \tilde{g}_{w} \frac{X (K_x + 1 + B) \tilde{T}_{w}}{\alpha}\right)
\]

The discretized form of this temperature can be expressed as:

\[
\frac{T_{w_{i+1,j}} - T_{w_{i,j}}}{\Delta U} = -\chi \frac{K}{w} \tilde{T}_{d_{i+1,j}} + \frac{K}{w} \tilde{T}_{w_{i+1,j}} + \frac{K}{w} \tilde{g}_{w_{i+1,j}} \frac{X (K_x + 1 + B) \tilde{T}_{w_{i,j}}}{2}
\]

Where \( w \) is the water heat capacity ratio:

\[
w = \frac{m_w c_w}{m_d c_p d}
\]

Equations (9), (12), (18) and (20) describe perfectly the heat and mass transfer between the counter flow air passages and water film.

Governing equations are subject to the following boundary and initial conditions:

\[
g_d(t,0) = g_{d,in}(t)
\]

\[
T_d(t,0) = T_{d,in}(t)
\]

\[
g_w(0,x) = g_w(t_i, X_i - x)
\]

\[
T_d(0,x) = T_w(t_i, X_i - x)
\]

Finite-difference method [15] is extended for simulating the combined heat and mass transfer processes that occur in the regenerative single stage cooler.

### 2 Thermal Model of a Residential Building with Regenerative Evaporative Cooling System

Temperature profile for the regenerative single stage counter flow configuration are shown in Fig. 5. Wet and dry streams flow in opposite directions. The rate of heat transfer between the water film and primary air move inversely to the rate of evaporation. This occurs because the water temperature increase in the direction of flow of secondary airstream.

The product air temperature approaches the inlet wet bulb temperature, especially for low inlet relative humidity ratio. This is the ultimate temperature at the dry passage exit where the water and primary air temperature are almost
equal.

Fig. 5 Temperature Distribution inside the cooler:
\[ T_{\text{din}} = 34.2^\circ\text{C}, T_{\text{wfin}} = 22^\circ\text{C} \]

To validate the computational model, numerical results were compared with experimental data obtained by Hsu et al. [14]. For similar primary air temperature, humidity and mass flow rate, temperature distributions of air flowing in the cooler and water film compare well with the experimental results.

2.1 Evaluation on Evaporative Cooling System Performance

The cooling capacity of the Evaporative Cooling System is calculated by the difference between the inlet and product air temperature as follows:

\[
Q_c = m_d C_p d \left( T_{d,\text{in}} - T_{\text{wetb}} \right) - T_{d,\text{out}} \left( T_{d,\text{in}} - T_{\text{wetb}} \right) \tag{27}
\]

Another indicator describe the cooling performance which is determined by the air rate per unit cooling capacity [16]:

\[
E_m = \frac{m_p v}{Q_c} \tag{28}
\]

The performance of the cooler depend on various parameter including working conditions. For the initiali values ranges, shown in Table 1, the effect of the process air temperature and relative humidity is showin in Fig. 6 and Fig. 7.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Unit</th>
<th>Initial value</th>
<th>Ranges</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient air temperature</td>
<td>°C</td>
<td>35</td>
<td>20-45</td>
</tr>
<tr>
<td>Ambient air Relative Humidity</td>
<td>%</td>
<td>40</td>
<td>30-80</td>
</tr>
<tr>
<td>Primary air flow rate</td>
<td>m/s</td>
<td>2.7</td>
<td>0.9-3</td>
</tr>
</tbody>
</table>

The room initial temperature is assumed to be 30 °C and relative humidity 50%. The results shows that QC values drops from 32.64 W/m² to 175.2 W/m² and \( E_m \) drops from 0.9805 to 0.18 when the inlet tempreature increase from 20 °C to 45 °C.

For the specified range of the inlet primary air temperature, the cooling capacity of the system reflect the high performance achieved at high ambient temperature. The load of the room is dominated by the latent load which is shown by the steady decline of \( E_m \).

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For the specified range of the inlet primary air temperature, the cooling capacity of the system reflect the high performance achieved at high ambient temperature. The load of the room is dominated by the latent load which is shown by the steady decline of \( E_m \).

Inversely, the QC values drops from 185 W/m² to 36.49 W/m² and \( E_m \) increases from 0.173 to 0.87 when the inlet tempreature relative
humidity increases from 30% to 80%.

These results show that for extreme values of air humidity, the performance of the cooler will deteriorate rapidly. So an ideally designed system would achieve adequate performances for high temperature and comfortable range of relative humidity 30%-60%.

### 2.2 Thermal Model of a Residential Building

Thermal model of a generic house, with specified geometry and material thermal properties, is used to evaluate the cooling cost for the proposed cooler at optimum performance and working conditions.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>House geometry</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>House surface</td>
<td>200</td>
<td>m²</td>
</tr>
<tr>
<td>House height</td>
<td>3.7</td>
<td>m</td>
</tr>
<tr>
<td>Number of windows</td>
<td>2-6</td>
<td></td>
</tr>
<tr>
<td>Width of window</td>
<td>1.2</td>
<td>m</td>
</tr>
<tr>
<td><strong>House insulation</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wall wool thickness</td>
<td>0.22</td>
<td>m</td>
</tr>
<tr>
<td>Glass window thickness</td>
<td>0.0095</td>
<td>m</td>
</tr>
</tbody>
</table>

To evaluate the thermal performances of a building using integrated regenerative single stage cooler, the following assumptions are considered:

- An initial temperature of the house of 30°C
- An initial indoor air temperature of 40°C
- An initial indoor air relative humidity of 40%
- The cost of electricity is 0.08$ per kilowatt/hour
- All electric energy is transformed to heat energy.

The outdoor environment is modeled with an infinite heat capacity varying with \( \Delta T_{\text{outdoor}} = 5°C \) to approximate daily temperature fluctuation. The regenerative evaporative cooler heat flow and heat losses to the environment are considered to calculate the residential building temperature variations, as shown in Fig. 8.

The temperature of the building is affected by many parameters such as the room surface, the wall material and thickness and the type of insulation. As the building temperature is related to the cooling cost, these parameters should be carefully fixed.

Considered a typical outdoor summer environment, with recommended relative humidity range and optimum cooler performances, the variation of indoor temperature and corresponding cooling cost is observed for various numbers of building windows as below.
The number of windows in the occupied space have a large influence in the home comfort. If the number of windows are reduced from $n_w=6$ to $n_w=2$, the indoor temperature drops from 28.66 °C to 24.91 °C and the cooling cost is reduced from 85.84 $ to 58.65$ for around 6 days.

4 Conclusion

Single stage regenerative evaporative cooling system has been presented, then, thermal model of a generic residential building with the cooler operating in optimum performances has been proposed in the current effort.

Simulation were carried out with heat and mass transfer studies. Numerical results suggests that the proposed Evaporative Cooling system is capable of cooling air to temperatures lower than the ambient wet bulb temperature insuring a considerable cooling capacity and high efficiency. In addition, the thermal model of the typical residential house was found to evaluate the cooling performance and the cost effectiveness for various condition and parameters. Additional parametric studies and experiments are needed to better understand the influence of different parameters, including the size of the cooler and to improve its performances. Numerical and experimental studies of the proposed system with modified heat exchanger tool id underway.

Nomenclature

- $m_d$: Air mass flow rate in the dry passage
- $m_a$: Air mass flow rate in the wet passage
- $m_p$: Product Air mass flow rate
- $c_{pd}$: Specific heat capacity of the air in the dry passage
- $c_{pw}$: Specific heat capacity of the air in the wet passage
- $T_d$: Air temperature in the dry passage
- $T_{wf}$: Water film temperature
- $T_w$: Air temperature in the wet passage
- $k$: Overall heat transfer coefficient
- $A$: Cooler Passage depth
- $l$: Cooler Passage length
- $V$: Room volume
- $h_w$: Air enthalpy in the wet passage
- $h_{wv}$: Air vapor enthalpy in the wet passage
- $\beta$: Mass transfer coefficient
- $g_d$: Air moisture content in the dry passage
- $g_{wf}$: Saturated air moisture content
- $g_w$: Air moisture content in the wet passage
- $\alpha$: Convective heat transfer coefficient
- $Nu$: Nusselt number
- $k_a$: Air thermal conductivity
- $d_h$: Hydraulic diameter of the wet channel
- $Le$: Lewis number
- $r_{wb}$: Heat of vaporization of water at wet bulb point value
- $t$: Time
- $T$: Temperature
- $in$: At the inlet of the passage
- $out$: At the outlet of the passage
- $wetb$: Wet bulb

References:


