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.

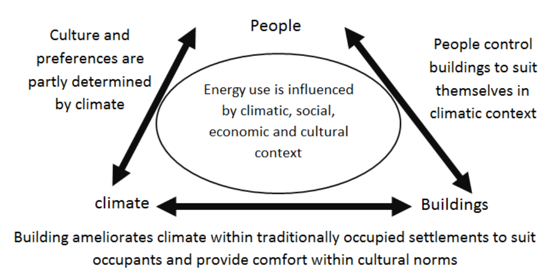

Fig. 1 Energy needs traditional three-way interaction (source: Nicol et al, 2012 [3])

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.

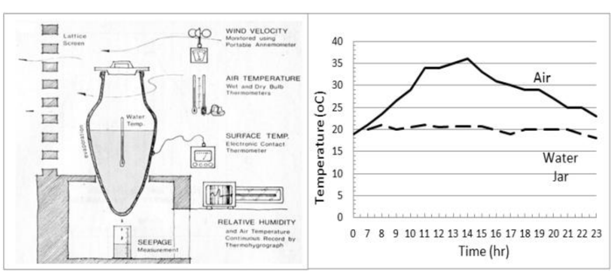

Fig. 2 Traditional passive evaporative cooling : Maziara water jar

Over 16 hour test period, a single jar produced 1700 Kcal of cooling. And the cooling rate reached 192 watts.

The organization of this article is as follows. In Sec. 2. a passive regenerative evaporative cooling system is presented. Following that, the equation of heat and mass balance are developed in Sec.3. Subsequently, a thermal model providing the cooling cost of a typical residential building is proposed in Sec. 4. Concluding remarks are collected together in Sec. 5

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. . 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.

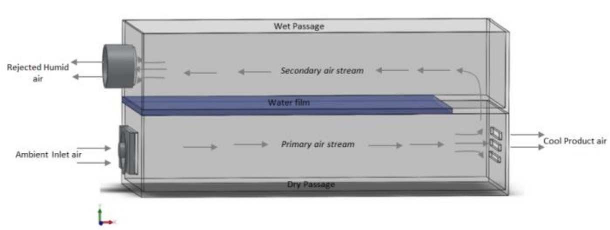

Fig. Regenerative sub-wet-bulb temperature IEC operation mode

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.

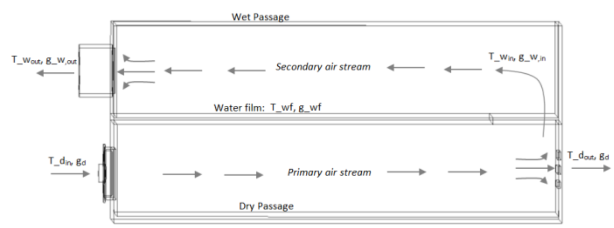

Fig. 3 Schematic of the IEC system

Basic assumptions are considered:

- Water mass flow rate is constant,
- The cooler is assumed to be well insulated from its surroundings,
- The heat and mass transfer coefficients and Lewis factor inside each passage are constants,
- The specific heat of fluid, air and vapor are constants,
- Humidity ratio of air in equilibrium with water surface is assumed to be a linear function of the water surface temperature.
- The process is steady-state

Under these assumptions, the energy conservation balance is written as

$$\frac{m_d C_{pd}}{A}\frac{\partial T_d}{\partial x} = -K_s(T_d - T_{wf}) \quad (1)$$

$$\frac{m_w}{A}\frac{\partial h_w}{\partial x} = h_{wv}\,\beta(g_{wf} - g_w) + \alpha(T_{wf} - T_w) \quad (2)$$

The convective heat transfer coefficient between the secondary air stream and the water film is given by the Nusselt Number

$$Nu = \alpha \frac{d_h}{k_a} \tag{3}$$

Similarly, the mass transfer coefficient, β in is given by the Lewis number correlation

$$Le = \frac{\alpha}{\beta c_p} \tag{4}$$

The conservation of mass equation is expressed as

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

The energy balance at the water film interface is given by

$$\frac{m_w c_{pw}}{D}\frac{\partial T_{wf}}{\partial x} = U(T_w - T_{wf}) - \beta h_{vw}(g_{wf} - g_w) - \alpha(T_{wf} - T_w) \tag{6}$$

2.1.1 Nondimensionalization

With the transformation $\tilde{T} = (T - T_{wb})\tau$,where $\tau = {}^{1}/_{(T_{in} - T_{wb})}$, and $\tilde{g} = (g - g_{wb})\sigma$,where $\sigma = {}^{1}/_{(T_{in} - T_{wb})}$, the passages temperature and moisture content may be written in terms of nondimensional parameters as [13]

$$\widetilde{T_d} = \frac{T\_d - T_{wb}}{T_{d,in} - T_{wb}}, \quad \widetilde{T_w} = \frac{T_w - T_{wb}}{T_{w,in} - T_{wb}}, \quad \widetilde{T_{wf}} = \frac{T_{wf} - T_{wb}}{T_{d,i} - T_{wb}} \tag{7}$$

Similarly, the nondimensional moisture content ratios are

$$\widetilde{g_w} = \frac{g_w - g_{wb}}{g_{wb} - g_{d,i}}, \quad \widetilde{g_{wf}} = \frac{g_{wf} - g_{wb}}{g_{wb} - g_{d,i}} \tag{8}$$

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.

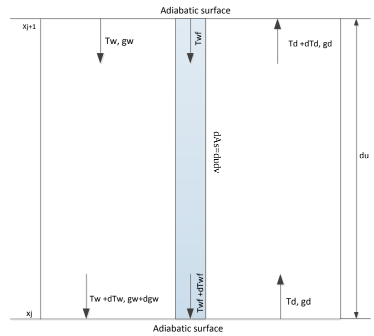

Fig. 4 One dimensional finite volume of the cooler

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

$$\frac{\partial \widetilde{T_d}}{\partial u_d} = -K\,\chi\,C\left(\check{T}_d - \check{T}_{wf}\right) \tag{9}$$

$$\frac{\check{T}_{d_{i+1,j}} - \check{T}_{d_{i,j}}}{\Delta U} = -KC\chi\frac{\check{T}_{d_{i+1,j}} + \check{T}_{d_{i,j}}}{2} + KC\chi\frac{\check{T}_{wf_{i+1,j}} + \check{T}_{wf_{i,j}}}{2} \tag{10}$$

With nondimensional coefficients:

$$K = \frac{K_s}{\alpha}, \qquad C = \frac{m_a c_{pa}}{m_d c_{pd}}, \qquad \chi = \frac{\alpha A_s}{m_a c_{pa}} \tag{11}$$

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

$$\frac{\partial \breve{g}_w}{\partial u_w} = -\frac{L_e \alpha A_s}{m_a c_{pa}}\breve{g}_w + L_e^2 B\frac{L_e \alpha A_s}{m_a c_{pa}}\check{T}_{wf} \tag{12}$$

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

$$\breve{g}_w = a + b\check{T}_w \quad and \quad \breve{g}_{wb} = a + b\check{T}_{wb} \tag{13}$$

Where b is obtained as

$$b = \frac{\breve{g}_w - \breve{g}_{wb}}{\check{T}_w - \check{T}_{wb}} \tag{14}$$

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

$$\breve{g}_w = \frac{\beta\, c_{pa}}{\alpha}\frac{br_{wb}}{c_{pa}}\check{T}_w = Le\; B\; T_w \tag{15}$$

Assuming that r_{wb} and c_{pa} are almost constant, the magnitude of B depend on b which depends on both $\check{T}_w$ and $\check{T}_{wb}$ [13].

Transforming equation (12) into the discretised form:

$$\frac{\check{g}_{w}-\check{g}_{w_{i,j}}}{\Delta U}= -L_e\chi\frac{\check{g}_{w_{i+1,j}}+\check{g}_{w_{i,j}}}{2}-L_e^2 B\chi\frac{T_{wf_{i+1,j}}+T_{wf_{i,j}}}{2} \quad (16)$$

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

$$\frac{\partial h_w}{\partial u_w}=(c_{pw}+g_w c_{pv})\frac{\partial T_w}{\partial u_w}+(r_{w0}+c_{pv}T_w)\ \frac{\partial g_w}{\partial u_w} \quad (17)$$

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:

$$\frac{\partial T_w}{\partial u_w}= -\chi\left(T_w-T_{wf}\right) \quad (18)$$

$$\frac{T_{w_{i+1,j}}-T_{w_{i,j}}}{\Delta U}=-\chi\frac{T_{w_{i+1,j}}+T_{w_{i,j}}}{2}+\chi\frac{T_{wf_{i+1,j}}+T_{wf_{i,j}}}{2} \quad (19)$$

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

$$\frac{\partial\check{T}_{wf}}{\partial u_w}=-\frac{\chi K}{w}\check{T}_d+\frac{\chi}{w}\check{T}_w+\frac{\chi}{w}\ \check{g}_w\ \frac{\chi}{w}(\frac{K_s}{\alpha}+1+B)\check{T}_{wf} \quad (20)$$

The discretized form of this temperature can be expressed as:

$$\frac{\check{T}_{wf_{i+1,j}}-\check{T}_{wf_{i,j}}}{\Delta U}=-\ \frac{\chi K}{w}\frac{\check{T}_{d_{i+1,j}}+\check{T}_{d_{i,j}}}{2}+\frac{\chi}{w}\frac{\check{T}_{w_{i+1,j}}+\check{T}_{w_{i,j}}}{2}+\frac{\chi}{w}\frac{\check{g}_{w}+\check{g}_{w_{i,j}}}{2}-\ \frac{\chi}{w}(\frac{K}{\alpha}+1+B)\ \frac{\check{T}_{wf_{i+1,j}}+\check{T}_{wf_{i,j}}}{2} \quad (21)$$

Where w is the water heat capacity ratio:

$$w=\frac{m_w c_w}{m_d c_{pd}} \quad (22)$$

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:

$$g_d(t,0)=g_{d,in}(t) \quad (23)$$

$$T_d(t,0)=T_{d,in}(t) \quad (24)$$

$$g_w(0,x)=g_w(t_l,X_l-x) \quad (25)$$

$$T_d(0,x)=T_w(t_l.X_l-x) \quad (26)$$

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

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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 inversly to the rate of evaporation. This occurs because the water temperature increase in the direction of flow of secondary airstream.

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

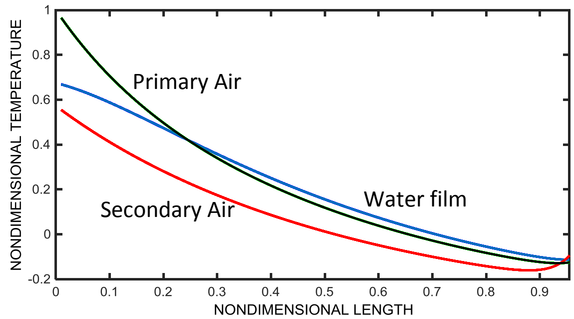

Fig. 5 Temperature Distribution inside the cooler: $T_{din} = 34.2$℃,$T_{wfin} = 22$℃

To validate the computational model, numerical results were compared with experimental data obtained by Hsu et al. [14]. For similar primary air temeprature, humidity and mass flow rate, temperature distrubutions 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 diffreence between the inlet and product air temperature as follows:

$$Q_c = m_d Cp_d \left[T_{d,in}\left(T_{d,in} - T_{wetb}\right) - T_{d,out}\left(T_{d,in} - T_{wetb}\right)\right] \quad (27)$$

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

$$E_m = \frac{m_{p\,V}}{Q_c} \quad (28)$$

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 1 Evaluation parameters

Parameters	Unit	Initial value	Ranges
Ambient air temperature	°C	35	20-45
Ambient air Relative Humidity	%	40	30-80
Primary air flow rate	m/s	2.7	0.9-3

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/m^2 to 175.2 w/m^2 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 refelect 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.

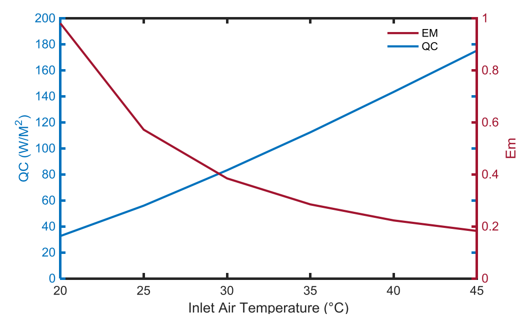

Fig. 6 Cycle performance variation with the inlet process air temperature

Inversely, the QC values drops from 185 w/m^2 to 36.49 w/m^2 and E_m increases from 0.173 to 0.87 when the inlet tempreature relative humidity increases from 30% to 80%.

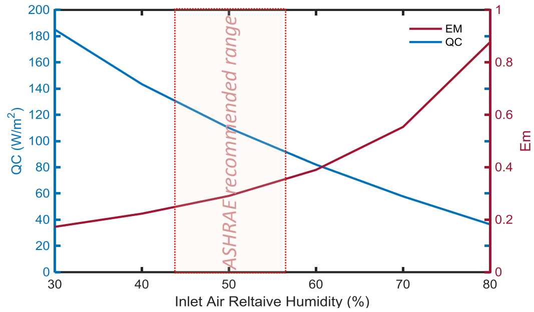

Fig. 7 Cycle performance variation with the inlet process air Relative Humidity

These results shows that for extreme values of air humidity, the performance of the cooler will deterioate rapidly. So an ideally designed system would achieve adequate performances for high temeperature and confortable range of relative humidity 30%-60%[1].

2.2 Thermal Model of a Residential Building

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

Table 2 Residential building thermal performances

Parameter	Value	Unit
House geometry		
House surface	200	m^2
House height	3.7	m
Number of windows	2-6	-
Width of window	1.2	m
House insulation		
Wall wool thickness	0.22	m
Glass window thickness	0.0095	m

[1] To manage health effects and illness, ASHRAE suggests a range of 45%-55%.

To evalaute the thermal performances of a building using integarted regenerative single stage cooler, the following assumption 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 $\Delta T_{outdoor} = 5°C$ to approximate daily temperature fluctuation.
The regenerative evaporative cooler heat flow and heat losses to the environmeent are considered to calculate the residential building temperature varaiations, as shown in Fig. 8.

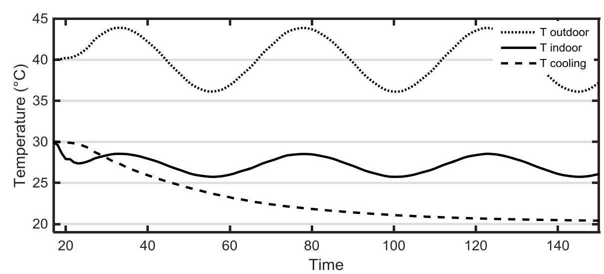

Fig. 8 Generic house temperature variation with typical process air temperature and humidity

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

Considerenig a typical outdoor summer environment, with recommanded relative humidity range and optimum cooler performances, the varaiation of indoor temperature and corresponding cooling cost is observed for various number of building windows as bellow.

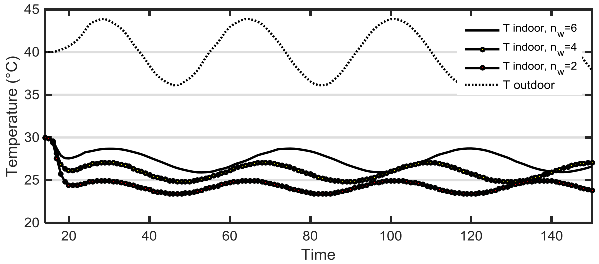

Fig. 9 The effect of building windows number on the indoor cooling performances

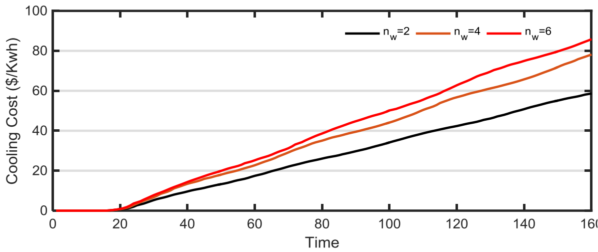

Fig. 10 Cooling cost of the residential building with various number of window

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 ℃ to 24.91 ℃ 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
Td	Air temperature in the dry passage
Twf	Water film temperature
Tw	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
β	Mass transfer coefficient
gd	Air moisture content in the dry passage
gwf	Saturated air moisture content
gw	Air moisture content in the wet passage
α	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:

[1] W.Johnson, Keeping Cool, *Saudi Armaco World*, Arab and Islamic cultures and connections, Vol.46, 1995, pp.10-17.

[2] J. Karlsson, Possibilities of using thermal mass in buildings to save energy, cut power consumption peaks and increase the thermal comfort, *Licentiate thesis, Lund Institute of*

Technology, Sweden, ISSN 0348-7911, 2012, pp.100.

[3] A. Sommers, Q. Wang b, X. Hanb, C. T.Joenc, Y. Parkd and A. Jacobi , *Ceramics and ceramic matrix composites for heat exchangers in advanced* thermal systems-A review, Applied Thermal Engineering, Elsevier, 2010, pp. 1277-1291.

[4] D. Yogi Goswami and Y. Zhao, *Solar Energy and Human Settlements*, Proceedings of ISES World Congress, Springer science &Business Media , 2007, Vol.1-5.

[5] B.P. Ager and J.A. Tickner, *The control of microbiological hazards associated with air-conditioning and ventilation systems*, Oxford Journals, Life Sciences & Medicine, Vol. 27, Issue 4, 1983, pp. 341-358.

[6] A.Cain, F. Afshar, J. Nortin and M.R. Daraie, *Traditional Cooling Systems in the Third World*, Echologist, vol.6, 1976, pp. 61-67.

[7] X. Zhao, JM. Li, SB. Riffat, *Numerical study of a novel counter-flow heat and mass exchanger for dew point evaporative cooling*, Applied Thermal Engineering, 2008, pp.1942-1951.

[8] D. R. Crum, J. W. Mitchell, W. A. Beckman, *Indirect evaporative cooler performance*, ASHRAE transactions, 1987, pp.1261-1275.

[9] G. Boxem, S. Boink, W .Zeiler, *Performance model for small scale indirect evaporative cooler*, Proceedings of the 9th REHVA World Congress: WellBeing Indoors, Finland, 2007, ISBN 978-952-99898-2-9.

[10] X. Zhao, JM. Li, SB. Riffat, *Numerical study of a novel counter-flow heat and mass exchanger for dew point evaporative cooling*, Applied Thermal Engineering, 2008, pp.1942-1951.

[11] X. Zhao, Z. Duan, C. Zhan, SB. Riffat, *Dynamic performance of a novel dew point air conditioning for the UK buildings*, International Journal of Low-Carbon Technologies, 2009, pp. 27-35.

[12] A.Hasan, M.Vuolle, K .Sirén, R.Holopainen P. A Tuomaala, *Cooling Tower Combined With Chilled Ceiling- System Optimisation*, International Journal of Low Carbon Technologies, 2007, pp 217-224.

[13] B. Halasz, *A general mathematical model of evaporative cooling devices*, Rev Gen Therm Journal, Elsevier, vol. 37, 1998, pp.245-255.

[14] S. Hsu, Z. Lavan, W. Worek, *Optimization of wet surface heat exchangers*, Energy Vol. 14, No. 11, 1989, pp. 757-770.

[15] W. Zheng, W.M. Worek, A. Part, *Numerical simulation of combined heat and mass transfer process in a rotary dehumidifier*, Numerical Heat Transfer 23 (1993), 1993, pp. 211–232.

[16] W. Gao, W. Worek, V. Konduru, K.Adensin,
Numerical study on performance of a desiccant cooling system with indirect evaporative cooler, Energy and Buildings, Vol. 128, 2016, pp. 834-844.