Univerzitet u Beogradu
Poljoprivredni fakultet
Institut za poljoprivrednu tehniku
Naučni časopis
POLJOPRIVREDNA TEHNIKA
Godina XXXVIII
Broj 3, 2013.
Strane: 27 – 39

University of Belgrade
Faculty of Agriculture
Institute of Agricultural Engineering
Scientific Journal
AGRICULTURAL ENGINEERING
Year XXXVIII
No. 3, 2013.
pp: 27 – 39

UDK: 536.7

Originalni naučni rad
Original scientific paper

A SIMPLE MODEL FOR EVAPORATIVE COOLING SYSTEM OF A STORAGE SPACE IN A TROPICAL CLIMATE

Chinenye Ndukwu[1*], Seth Manuwa[2], Olawale Olukunle[2], Babatunde Oluwalana[3]

[1]Michael Okpara University of Agriculture, College of Engineering and Engineering Technology, Department of Agricultural and Bio Resources Engineering, Umudike, Abia State, Nigeria
[2]Federal University of Technology Akure, School of Engineering and Engineering Technology, Department of Agricultural Engineering, Akure, Ondo State, Nigeria
[3]Federal University of Technology Akure, School of Agriculture and Agricultural Technology, Department of Food Science and Technology, Akure, Ondo State, Nigeria

Abstract: This paper deals with the development of a simple mathematical model for experimental validation of the performance of a small evaporative cooling system in a tropical climate. It also presented the coefficient of convective heat transfer of wide range of temperatures based on existing model. Extensive experiments have been performed during January to February 2013 for a small evaporative cooler designed for storage of fruits and vegetables. The model considered the thermal properties of the material of the cooling pad and assumed that the cooling pad is a plain porous wall bounded by two convective airs at different temperature at the two surfaces. The predicted and experimental value of various cooling efficiency at different range of inlet temperature has been determined. In addition the values of the coefficient of convective heat transfer for a wide range of temperatures is also presented.

Key words*: evaporative cooling, cooling pad, model equation, heat transfer*

[*] Corresponding author. Email: ndukwumcu@gmail.com

INTRODUCTION

The basic principle of evaporative cooling is cooling by evaporation. When water evaporates, it draws energy from its surroundings, which produces a considerable cooling effect. Evaporative cooling occurs when air, that is not too humid, passes over a wet surface (humidifier). The movement of the air can be passive i.e. when the air flows naturally through the pads or active with fans or blowers. The driving force for heat and mass transfer between air and water is the temperature and partial vapor pressure differences. Water is the working fluid in evaporative cooling thus it is environmentally friendly [1]. Due to the low humidity of the incoming air some of the water evaporates. This evaporation causes two favorable changes: a drop in the dry-bulb temperature and a rise in the relative humidity of the air. This non-saturated air cooled by heat and mass transfer is forced through enlarged liquid water surface area for evaporation by utilizing blowers or fans. Some of the sensible heat of the air is transferred to the water and becomes latent heat by evaporating some of the water. The latent heat follows the water vapor and diffuses into the air. In a DEC (direct evaporative cooling), the heat and mass transferred between air and water decreases the air dry bulb temperature (DBT) and increases its humidity, keeping the enthalpy constant (adiabatic cooling) in an ideal process. However, 100% saturation is impossible for direct evaporative coolers due to two reasons [2]. Firstly, most of the pads are loosely packed or with cells, therefore the process air can easily escape between the pads without sufficient contact with the water. After water evaporates, it enters the air as water vapor and conveys the heat absorbed during evaporation back to the air in the form of latent heat. The effectiveness of this system is defined as the rate between the real decrease of the DBT and the maximum theoretical decrease that the DBT could have if the cooling were 100% efficient and the outlet air were [1]. Practically, wet porous materials or pads provide a large water surface in which the air moisture contact is achieved and the pad is wetted by dripping water onto the upper edge of vertically mounted pads. According to [1] experimental studies are reliable and convincible; but they are usually costly and too tasking. In addition, the experiment results were obtained under various testing conditions which are affected by the environmental conditions with given inlet parameters and the results may be different when testing conditions is changed. Modeling analysis of evaporative cooling system is essential to explain the heat and mass transfer process in evaporative cooling and to predict the process outputs at various conditions. Over the years a number of models have been developed to describe direct evaporative cooling systems, not supported with a heat exchanger. [3, 4, 5, 6]

Most of these models ignore the thermal properties of the cooling pad material which will definitely affect the temperature drop inside the cooling chamber no matter how small. The models consider the heat and mass transfer that occur at the surface of the pad but in actual fact the cooling pad is a plain porous wall with thickness. Also the heat and mass transfer is not only on the surface but across the thickness with the air at the outer surface at different temperature from the air at the inner surface. The paper presents a simple model for direct evaporative cooling, incorporating the thermal properties and the thickness of the material of the pad in the heat and mass transfer process in a tropical environment. It also presents wide range of heat transfer coefficient at the tested conditions.

MATERIAL AND METHODS

Basic Mathematical Model

Model assumptions

1. The system is one dimensional.
2. The system is adiabatic.
3. The inside and the outside temperature of the air is different.
4. The two surfaces of the cooling pad are at different temperatures.
5. The cooling pad is surrounded by air at the two surfaces.
6. The heat transfer coefficient is different for the ambient air and the air bounding the cooling pad on the inside the cooler.
7. The cooling pad is a plain porous wall bounded by two convective fluids (air) at different temperatures.
8. The surface of the pad is completely wet.
9. The pad and the cold air inside are at the same temperature.
10. The water inlet and outlet temperature is the same.
11. The cooling pad is rigid.

Basic model equation

On the assumption that the cooling pad is a plain porous wall bounded by two convective fluids (air) outside the pad surface and inside the cooler, each at different temperature, the elementary sensible heat flux in terms of overall temperature and thermal properties of the pad for Fig. 1 is given by:

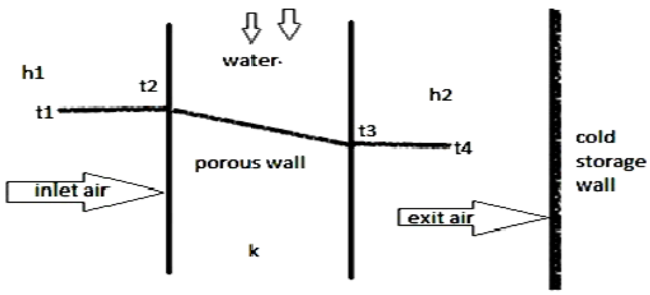

Figure 1. Scheme of the heat transfer process across the porous evaporative cooling pad

(1)

where:

q	[W·m^{-2}]	- heat flux,
h_1	[W·m^{-2}·K^{-1}]	- convective heat transfer coefficient of the outside air,
T_1	[°C]	- outside air temperature,
T_2	[°C]	- inlet temperature to the porous pad,
A	[m^2]	- surface area of the pad.

$$dq = \left(\frac{T_2 - T_3}{\frac{x_{23}}{k_{23}}}\right) \mathrm{dA} \tag{2}$$

where:

x_{23} [m] - thickness of the porous pad,
k_{23} [W·m^{-1}·K^{-1}] - thermal conductivity of the pad material,
T_2 [°C] - inlet air temperature into the porous pad,
T_3 [°C] - exit air temperature from the porous pad,
A [m^2] - surface area of the pad.

$$dq = h_2 A(T_3 - T_4)dA \tag{3}$$

where:

h_2 [W·m^{-2}·K^{-1}] - convective heat transfer coefficient of the inside air,
T_3 [°C] - exit air temperature from the porous pad,
T_4 [°C] - inside air temperature of the evaporative space.

The heat balance for the three equations gave:

$$dq = \left(\frac{T_1 - T_4}{\frac{1}{h_1} + \frac{x_{23}}{k_{23}} + \frac{1}{h_2}}\right)dA \tag{4}$$

[7] gave the mass flow rate of re-circulating water evaporating into air from a surface in terms of the mass transfer coefficient for evaporative cooler as:

$$\acute{\omega} + \left(\frac{\delta\acute{\omega}}{\delta A}\right)dA = \acute{\omega} + h_D(\omega_s - \omega)dA \tag{5}$$

where:

$\acute{\omega}$ [kg·h^{-1}] - mass flow rate of water,
h_D [kg·m^{-}2·s^{-1}] - convective mass transfer coefficient,
ω [kg·kg^{-1}] - humidity ratio,
ω_s [kg·kg^{-1}] - moist air specific humidity.

He gave the simplified solution as:

$$d\acute{\omega} = h_D(\omega_s - \omega)dA \tag{6}$$

Based on equation 6 [6] stated that the water mass flow rate does not remain constant due to the process of evaporation. Simultaneous heat and mass transfer takes place at the air–water interface. [5] analyzed the interface of air – liquid of direct evaporative cooling system, by energy conservation and gave the heat passing through the air -water interface as:

$$dq = m_a c_{pu} dT \tag{7}$$

where:

m_a [kg·h^{-1}] - mass flow rate of air,

C_{pu} [kJ·kg^{-1}·K^{-1}] - humid specific heat.

Were C_{pu} is given by:

$$c_{pu} = c_{pa} + wc_{pw} \tag{8}$$

where:
c_{pa} [kJ·kg^{-1}·K^{-1}] - specific heat of dry air,
c_{pv} [kJ·kg^{-1}·K^{-1}] - specific heat of vapor.

Since the quantity of heat loss by draft air and the cooling pad is equal to heat passing through air- water interface. The overall energy balance on the process fluid and the cooling pad will be:

$$\left(\frac{T_1 - T_4}{\frac{1}{h_1} + \frac{x_{23}}{k_{23}} + \frac{1}{h_2}}\right) dA = m_a c_{pu} dT \tag{9}$$

The above equation can be integrated resulting in:

$$\frac{1}{m_a c_{pu}\left(\frac{1}{h_1} + \frac{x_{23}}{k_{23}} + \frac{1}{h_2}\right)} \int_0^A dA = \int_{T_1}^{T4} \frac{dT}{T_1 - T_4} \tag{10}$$

The integration will yield:

$$-\left[\frac{A}{m_a c_{pu}\left(\frac{1}{h_1} + \frac{x_{23}}{k_{23}} + \frac{1}{h_2}\right)}\right] = in\left[1 - \frac{T_1 - T_4}{T_1 - T_w}\right] \tag{11}$$

(T_1-T_4/T_1-T_W) is the evaporative efficiency or effectiveness and is represented by $\mathcal{E}$.
With the hypothesis that air and vapor are perfect gases, [5] gave the enthalpy change as:

$$h_g - h = C_{pu}(T_s - T) + h_{vs}(w_s - w) \tag{12}$$

were:
h_{vs} [kJ·kg^{-1}] - vapour enthalpy of water at the surface temperature.

Assuming that $h_{vs} \approx h_{lvs}$ the above equation becomes:

$$h_g - h = C_{pu}(T_s - T) + h_{ls}(w_s - w) \tag{13}$$

were:
h_{lvs} [kJ·kg^{-1}] - specific enthalpy of water at the surface temperature of the cooling pad,
h_g [kJ·kg^{-1}] - specific enthalpy of saturated water vapour.

However in the presence of difference in enthalpy the term $h_{ls}(w_s-w)$ is neglected [8] therefore the change in enthalpy for the direct evaporative cooler can be written as:

$$\Delta H = c_{pu}\Delta T \tag{14}$$

were:
ΔH [kJ·kg^{-1}] - change in enthalpy,
ΔT [°C] - temperature difference.

Therefore:

$$\frac{\Delta H}{\Delta T} = c_{pu} \tag{15}$$

$$A = w \times L \tag{16}$$

were:
W [m] - width of the pad,
L [m] - length of the pad.

Therefore:

$$-\left[\frac{wL}{\frac{\Delta H m_a}{\Delta T}\left(\frac{1}{h_1}+\frac{x_{23}}{k_{23}}+\frac{1}{h_2}\right)}\right] = in[1-\mathcal{E}] \tag{17}$$

$$\mathcal{E} = 1 - exp\left\{-\left[\frac{wL}{\frac{\Delta H m_a}{\Delta T}\left(\frac{1}{h_1}+\frac{x_{23}}{k_{23}}+\frac{1}{h_2}\right)}\right]\right\} \tag{18}$$

The convective heat transfer is calculated from the Nusselt number as follows:

$$N_U = \frac{hl}{k} \tag{19}$$

Where k is the thermal conductivity of air, l is the characteristic length and is given by [5] as:

$$l = \frac{\ell}{A} \tag{20}$$

where:
ℓ [m^3] - volume occupied by the pad.

[9] gave a correlation to determine the heat transfer coefficient for a rigid evaporative media as:

$$N_u = 0.1\left(\frac{l}{x}\right)^{0.12} {R_e}^{0.8} Pr^{\frac{1}{3}} \tag{21}$$

where:
x [m] - thickness of the pad,
R_e [-] - Reynolds number,
N_u [-] - Nusselt number,
P_r [-] - Prandtl number.

$$R_e = \frac{vl}{\nu} \tag{22}$$

were:
v [$m \cdot s^{-1}$] - air speed,
ν [$m^2 \cdot s^{-1}$] - kinematic viscosity.

Also, the Prandtl number P_r is given by:

$$Pr = \frac{\nu}{\alpha} \tag{23}$$

Where α [$m^2 \cdot s^{-1}$] is the thermal diffusivity which is given by:

$$\alpha = \frac{k}{\rho c_{pa}} \tag{24}$$

where:
ρ [$kg \cdot m^{-3}$] - density of air.

The mass flow rate was generated from the continuity equation as follows:

$$m_{a= \rho A_1 v} \tag{25}$$

were A_1 is the area of the pad covered by each of the three fan since the pad is divided into three compartments.

Experimental Tests

An experimental test was conducted with palm fruit fiber as the cooling pad material at inlet air velocities of 4.0 $m \cdot s^{-1}$ and exit speed of 1.6 $m \cdot s^{-1}$. The characteristic length of the pad is 0.3 with a height of 1 m. The pad was divided into three equal area with each mounted with an axial fan delivering 0.5 $kg \cdot s^{-1}$ of air at a pad face velocity of 1.6 $m \cdot s^{-1}$. The test facility figure 2 was located under an open shade built under a whistling pine tree. This is to reduce direct action of the sun and expose the cooler to natural air. The test was carried out in January and February of 2013; this period presented the extremes of the temperature within the year. At this period, there was rain for 8 days, which presented very high ambient relative humidity of 80 %. In addition, the period presented extreme low humidity of 28 % and very high temperature of 45°C. The palm fruit fiber was loaded into the pad holder at a thickness of 30 mm and a parking density of 20 - 22 $kg \cdot m^{-3}$. The upper water tank delivers water at a rate of 10 $cm^3 \cdot s^{-1}$. The water flows through the pad by gravity into the bottom tank and re-circulates back with the water pump. The cooler is loaded with 2 kg of pumpkin (cucurbita) and amaranthus. A thermocouple (omega data logger, HH1147) (±0.1°C) was positioned through the hot wire terminals inserted into the cooling chamber. One of the terminals was covered with cotton wool soaked inside the water to measure the wet bulb temperature [10]. The air speed of the fan was determined with vane microprocessor (AM-4826) digital anemometer (±0.1 $m \cdot s^{-1}$). Two ABS temperature and humidity clock (±0.1°C and 1.0 %) was positioned inside the shade and another outdoor where there is no shade to record the temperature and humidity of the ambient. Two analogue thermometers were inserted inside the two tanks to measure the water temperature. The data were logged every two hours. The relative humidity and the enthalpy of the cooler were obtained from the psychometric chart. In addition, the wet bulb temperature and the enthalpy of inside the shade and the ambient were calculated also from the psychometric chart.

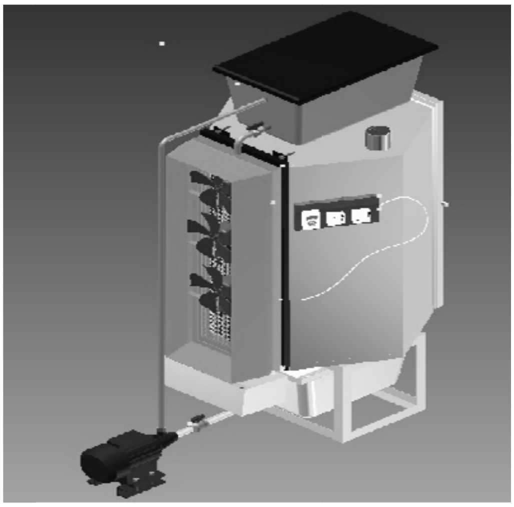

Figure 2. Evaporative cooling system test rig

Performance Evaluation

The cooling efficiency defined by Equation 25 is a widely used index for evaluating the performance of direct evaporative cooling systems [11, 12]. They were used as follows:

$$cooling\ efficiency(\%) = \frac{T_{db} - T_s}{T_{db} - T_w} \tag{26}$$

where:
T_{db} [°C] - dry bulb temperature of the outside air,
T_s [°C] - dry bulb temperature of the inside air,
T_w [°C] - wet bulb temperature of the outside air.

RESULTS AND DISCUSSION

Model validation

The mathematical model was validated using data generated from an existing active evaporative cooler Figure 2. The model validation was done at a constant draft air mass flow rate of 0.5 kg·s^{-1} and exiting air speed of 1.6 m·s^{-1} at a constant pad thickness of 30 mm and water flow rate of 10 cm^{3}·s^{-1} for the first day the ambient temperature ranged from 29.9-34.8°C with a relative humidity of 34-51% while it ranged from 26.1-34°C for the second day with a relative humidity of 38 -69%. Also at the third day the ambient temperature ranged from 27.8-34°C while the relative humidity ranged from 44-73%. At these prevailing environmental condition the cooler maintained a temperature range of 23.2-24.8°C with a relative humidity of 90.4 -94.8% on the first day while on the second day it provided a cooler temperature of 23.2-24.6°C with a relative humidity of 93.6-96.8%. On the third day the ambient condition provided a cooler temperature of 23.8-25.2°C with a relative humidity of 85.6-96.8%. The evaluation parameters (Table 1) were fitted into Eq. 18 to calculate the predicted cooling efficiency while the experimental efficiency was calculated with Eq. 27.

Also air properties determined from psychometric chart were fitted into Eq. 19-23 to calculate the coefficient of heat transfer for the two air conditions. Thermal conductivity of palm fiber was 0.057 W·m^{-20}K^{-1} [13]. The predicted and experimental cooling efficiency is presented in Fig. 3- 5.

Table 1. Evaluation parameters

Evaluation Parameters								
day one			*day two*			*day three*		
Time [hr]	*ΔH* [kJ/kg]*	*ΔT* [°C]*	*Time [hr]*	*ΔH [kJ/kg]*	*ΔT [°C]*	*Time [hr]*	*ΔH [kJ/kg]*	*ΔT [°C]*
10	2.05	4.0	10	2.90	6.0	9	1.54	2.9
12	0.40	6.5	11	6.58	6.7	11	2.37	4.8
14	5.05	9.7	12	4.99	8.2	13	2.04	7.4
16	0.18	8.2	13	5.08	9.0	15	2.12	9.7
18	4.51	7.7	16	3.42	10.0	17	6.74	8.7

* *ΔH* - change in enthalpy, *ΔT*- change in temperature.

In order to be certain there is difference in the predicted and experimental efficiency; analysis of variance was performed on the results of the three days of test. The *F* – value shows that there was no significant difference at the 5% level. The average mean efficiency difference on the first day for cooling efficiency and the predicted efficiency was 0.8% while the second day gave average mean difference of 3.45%. Also the third day gave an average mean difference of 3.72%. These values are very close to the experimental values as shown in Fig. 6.

This shows that the model have close to 96 – 99.2% accuracy. Figures 2 and 5 showed that the model performed relatively poorly at 16 hrs for the two days period compared to the rest of the period.

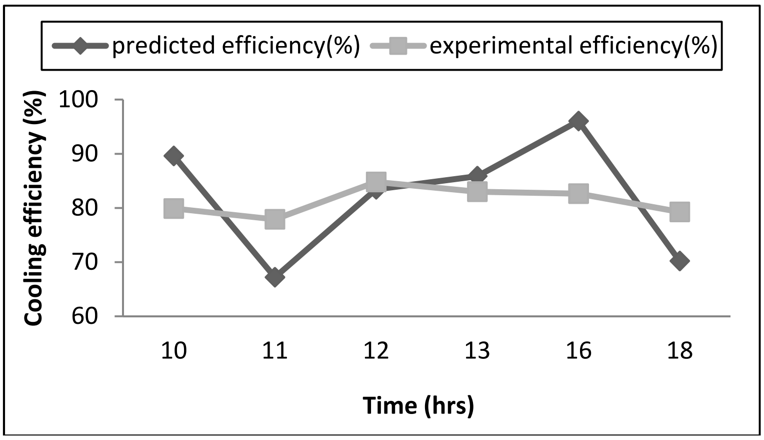

Figure 3. Hourly predicted and experimental cooling efficiency (%) for day one

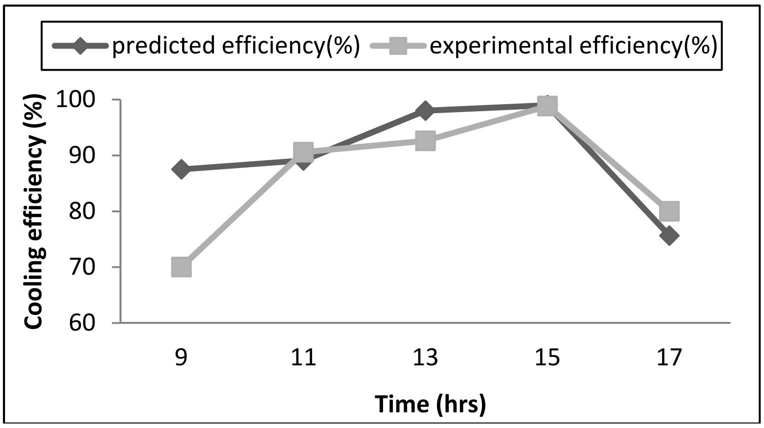

Figure 4. Hourly predicted and experimental cooling efficiency (%) for day two

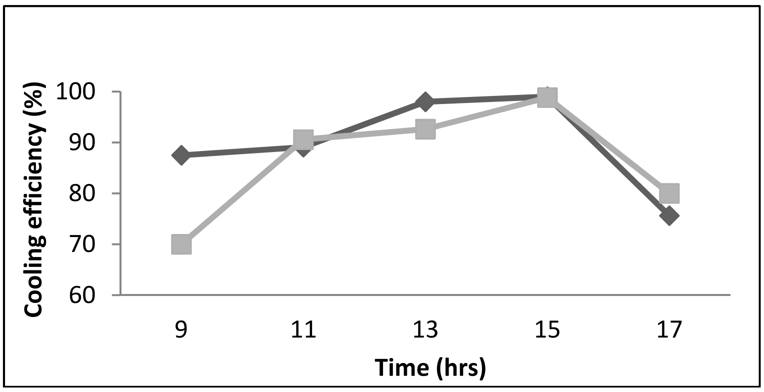

Figure 5. Hourly predicted and experimental cooling efficiency (%) for day three

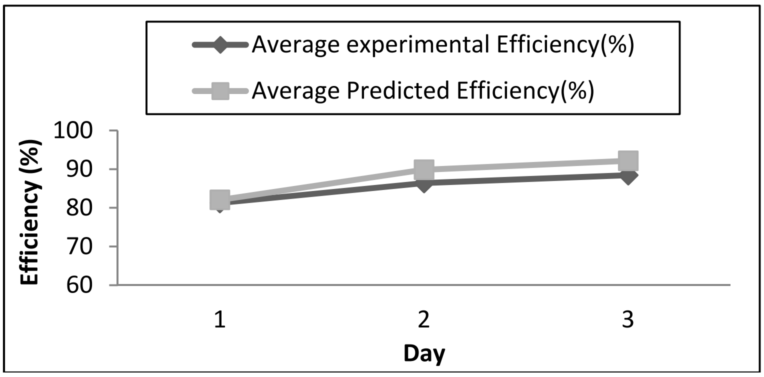

Figure 6. Average daily experimental and predicted efficiency

The best fit equation for the experimental and predicted value is given by:

$$pre(\mathcal{E}) = -2.745\exp{(\mathcal{E})^2} + 16.03\exp(\mathcal{E}) + 68.76 \quad (27)$$

where:

ε [%] - cooling efficiency.

The above equation has R^2 value of 100%.

The value of the Reynolds number at the inlet shows that the air is turbulent (Re > 5000) which is typical of force convection. Though the air enters into the cooling pad as a turbulent flow but it emerges as a laminar flow inside the cooling chamber (Re < 5000) as shown in Table 1. This shows that the cells of the cooling pad absorb some of the energy from the air.

CONCLUSION

The equation for predicting the cooling efficiency of a direct evaporative cooler for storage has been presented and validated. The model considered the thermal conductivity and thickness of the cooling pad material. The model has 96 – 99.2% accuracy. The plotting of the cooling efficiency relationship with time showed an exponential behavior. The results showed that the cooling efficiency of the systems gradually improves as the day goes by and peaks in the afternoon when the ambient temperature is highest in a tropical environment like Africa. At this period much cooling is required to maintain the cooler within the storage temperature, therefore the high efficiency. In the evening the ambient temperature decreases as shown from fig 3 - 5 and consequently the cooling efficiency decreases. The information presented could be useful in the design of evaporative cooling for other purposes.

REFERENCES

[1] Xuan, Y.M., Xiao, F., Niu, X.F., Huang, X., Wang, S.W. 2012. Research and application of evaporative cooling in China: A review (I). *Renewable and Sustainable Energy Reviews* Volume 16, Issue 5, p.p. 3535-3546.

[2] Di, Y.H., Liu, J.P., Huang, X. 2010. *Climate demarcating for application of evaporative cooling air conditioning* HVAC, 40 (2) pp. 108–111.

[3] Al-Helal, I., Short, T.H. 2003. A CFD study of naturally and ventilated greenhouses in extreme arid climates. ASAE Paper no. 99-5011 in Cooling Fan-ventilated Greenhouses: a Modelling Study. *Biosystems Engineering* 84 (3), p.p. 315–329.

[4] Liao, C., Chiu, K. 2002. Wind Tunnel Modeling the System Performance of Alternative Evaporative Cooling Pads in Taiwan Region. *Building and Environment*, No. 37, p.p. 177-187.

[5] Camargo, J.R., Ebinuma, C.D., Cardoso, S. 2003. A mathematical model for direct evaporative cooling air conditioning system. *Engenharia Térmica*, No. 4, p.p. 30-34.

[6] Qiang, T.W., Shen, H.G., Feng, J.M., Liang, R.J. 2006. Application of direct evaporative cooling air conditioning in moderate and high humid area of China. *Refrigeration and Air-conditioning*, 27 (3), pp. 21–25.

[7] Qureshi, B.A., Zubair, M.A. 2006. Comprehensive design and rating study of evaporative coolers and condensers. Part I. Performance evaluation. *International Journal of Refrigeration* 29, p.p. 645–658.

[8] El-dessouky, H.T., Ettouney, H.M., Bouhamra, W. 2000. A novel air conditioning system membrane air drying and evaporative cooling. *Trans IChemE,* Vol. 78, Part A, pp 999- 1009.

[9] Dowdy, J.A., Karabash, N.S. 1987. Experimental Determination of Heat and Mass Transfer Coefficients in rigid Impregnated Cellulose Evaporative Media. *ASHRAE Transactions*, part 2, v. 93., p.p. 382-395.

[10] Anyanwu, E.E. 2004. Design and measured performance of a porous evaporative cooler for preservation of fruits and vegetables. *Energy conversion and management* 45, p.p. 2187–2195.

[11] Jain. D. 2007. Development and testing of two-stage evaporative cooler. *Building and Environment*, 42, p.p. 2549–2554.

[12] Ndukwu, M. C. 2011. Development of Clay Evaporative Cooler for Fruits and Vegetables Preservation. *Agricultural Engineering International: CIGR Journal, 13*(1), p.p. 1-8.

[13] Krishpersad, M. 2012. Renewable building thermal insulation-oil palm fiber. *International journal of engineering and technology* 2, p.p. 475-479.

[14] Ditchfield, C., Tadini, C.C., Singh, R.K., Toledo, R.T. 2006. Velocity and temperature profile, heat transfer coefficient and residence time distribution of temperature dependent Herschel bukly fluid in tubular heat exchanger. *Journal of food engineering* 79, p.p. 632 -638.

MODEL EVAPORATIVNOG SISTEMA HLAĐENJA SKLADIŠNOG PROSTORA U TROPSKIM USLOVIMA

Chinenye Ndukwu*[1], Seth Manuwa[2], Olawale Olukunle[2], Babatunde Oluwalana[3]

[1]Poljoprivredni univerzitet Michael Okpara, Fakultet za inženjering i inženjersku tehnologiju, Institut za inženjering u poljoprivredi i bio-resursima, Umudike, Abia State, Nigeria
[2]Federalni tehnološki univerzitet Akure, Fakultet za inženjering i inženjersku tehnologiju, Institut za inženjering u poljoprivredi, Akure, Ondo State, Nigeria
[3]Federalni tehnološki univerzitet Akure, Fakultet za poljoprivredu i poljoprivrednu tehnologiju, Institut za hranu i prehrambenu tehnologiju, Akure, Ondo State, Nigeria

Sažetak: Ovaj rad se bavi razvojem jednostavnog matematičkog modela za eksperimentalnu ocenu performansi malog evaporativnog rashladnog sistema u tropskim klimatskim uslovima. Predstavljen je i koeficijent konvektivnog prenosa toplote velikog opsega temperature na osnovu postojećeg modela. Obimni eksperimenti su izvođeni tokom januara i februara 2013 sa malim evaporativnim uređajem za hlađenje skladišta za voće i povrće. Model je uzeo u razmatranje termičke osobine materijala rashladnog sloja i pretpostavio da je rashladni sloj ravan porozni zid ograničen sa dva konvektivna sloja vazduha na različitim temperaturama na obe granične površine. Određene su izračunate i eksperimentalne vrednosti različitih efikasnosti hlađenja pri različitim opsezima ulaznih temperature. Uz to, predstavljene su i vrednosti koeficijenata konvektivnog prenosa toplote za veliki opseg temperatura.

Ključne reči: *evaporativno hlađenje, rashladni sloj, jednačina modela, prenos toplote*

Prijavljen: *Submitted*:	02.07.2013.
Ispravljen: *Revised*:	20.09.2013.
Prihvaćen: *Accepted*:	21.09.2013.