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Mathematical Modeling of a Solar Passive Cooling System Using a Parabolic Concentrator

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Abstract

In this study a mathematical simulation of a system that is driven by a solar chimney equipped with Parabolic Solar Concentrator (PSC) in order to cool buildings by using an Evaporative Cooling Cavity (ECC). The Capability of the system to meet the standards and requirements has been considered in order for this study to be applicable in many fields (residential, industrial or agricultural). Mathematical modeling and simulation were performed to illustrate the effect of changing parameters regarding geometry and dimensions, ambient temperature, relative humidity, heat and mass transfer rate and others. The solar chimney has been designed by using parabolic solar concentrator to increase the efficiency of this system.The simulation showed that the system is capable of providing good temperature reduction even with low solar intensity (200 W/m²). It was found that the system could be used for residential and agricultural applications to provide thermal comfort conditions in hot air and dry climates.

Key words: Passive cooling; Parabolic solar concentrator; Evaporative; Air heater

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INTRODUCTION

Pollution is considered the main risk in the present century in which earth has been dealing with industrial, residential and agricultural wastes resulting from activities aiming to provide human comfort, and satisfaction.

Studies showed that all the creatures have a certain range of ecological comfort conditions (temperature, humidity, pressure and air quality). Such conditions were provided through systems derived directly or indirectly (electricity generation) by fossil fuels. Heating Ventilating and Air Conditioning (HVAC) systems are an example of such systems that depends totally on electricity to operate, which is usually generated by steam power plants that use fossil fuels to produce this electricity accompanied by emissions of Carbon Oxides, Sulfur Oxides, ash, soot and others. On the same side the HVAC system itself plays a major role in ozone depletion caused by chlorofluorocarbons (CFC's) and other kinds of refrigerants. Although HVAC system is yet the most efficient in providing the required conditions, still it has high initial and operating cost related to electricity consumption.

From here arose the responsibility of engineers to find a solution for all of the previous problems, a solution that ensures flexibility, acceptable efficiency and eco friendly operation.

Extensive research has been conducted on using evaporative cooling driven by natural ventilation for air conditioning applications, for example, Bahadori used a wind tower to capture the wind and then passing it through wetted conduit walls (Bahadori, 1985). Verma, Bansal and Garg (1986) studied the performance of apassive evaporative cooling system on the roof numerically. They concluded that this system significantly reduces air temperature of the room (Verma,). Giabaklou and Ballinger presented a method of passive cooling in low-rise multi-storey buildings through a simple water cascade combined with balconies and openings of individual units. In this system air passing over the water falling film to reduce ambient air temperature (Giabaklou & Ballinger, 1996). Aboulnaga used a roof solar chimney coupled with wind cooled cavity to optimize the performance of the system (Aboulnaga, 1998). Raman et al. studied two passive summer cooling models. The first one used two solar chimneys to ventilate the air and to act as an evaporative cooler during summer. The second model used a south wall as collector and a roof duct covered by a sack of cloth on the top as an evaporative cooling system. The thermal performance of the second model was found better than the first one (Raman, Mande, & Kishore, 2001). Manzan and Saro studied a passive system withan external air flows in ventilated roof with a wet lower surface cavity. They numerically investigated the thermal performance of the system by modeling of evaporative cooling process through the chimney (Manzan & Saro, 2002). Dai et al. presented a mathematical model of a new passive cooling system for humid climate using the solar chimney and adsorption cooling system. The system provides the cooling effect without increasing humidity of the room through increasing the rate of ventilation (Dai, Sumathy, Wang, & Li, 2003). Chungloo and Limmeechokchai investigated the performance of a passive cooling system under hot and humid climate experimentally. Their system equipped with a solar chimney and water spraying system that was placed on the roof. They reported that athigh ambient temperature the system performed well (Chungloo & Limmeechokchai, 2007).

Solar chimneys need high intensity of radiation in order to work as a draft sources, and to know that the global radiation in Jordan is one of the highest values worldwide, therefore, with this high yearly global radiation (2080 kWh/m²) and more than 300 sunny days annually (10 hours of daylight per one day) (Etier, Al-Tarabsheh, & Ababneh, 2010), Jordan maybe one of the most suitable places to use solar chimneys.

Passiveor natural cooling system can be employed as an alternative way to either maintain a cool and comfort home or reduce the load of the air-conditioning system which, save cooling energy consumption. In this study, the promising passive or natural cooling techniques are simulated using evaporative cooling cavity combined with natural day ventilation. If solar energy is available, a solar chimney is considered a good configuration to implement natural ventilation in buildings.

This study attempts to introduce a new low-energy passive technique to ventilate and provide thermal needs of occupants in the buildings using the evaporative cooling solar chimney system. In this work configuration, theory and improvements of the evaporative cooling cavity system equipped with a Parabolic Solar Concentrator solar chimney will be presented.

1 FLAT PLATE SOLAR CHIMNEY (SC)

The modeled solar system consists of two main parts: the solar chimney and the evaporative cooling cavity.The solar chimney is similar in principle and configuration to the solar collector. It consisted of a glass surface that works as a radiation transparent layer and an absorber wall that works as a heat intensifier. Both have the job of being capturing surfaces to the heat generated inside the chimney by the solar radiation. The solar chimney is oriented to the south and tilted with an angle equals 30° (Etier, Al-Tarabsheh, & Ababneh, 2010). The solar energy heated air in the SC, as a result air flows upward because of the stack effect which generate a draft that sucks the outside air through the system.

1.1 Modeling of the Solar Chimney

Figure 1 showed the element of the model for SC (Maerefat & Haghighi, 2010). In principle and based on the energy conservation law, a set of equations are obtained along the length of SC are described in details by Maerefat & Haghighi, 2010.

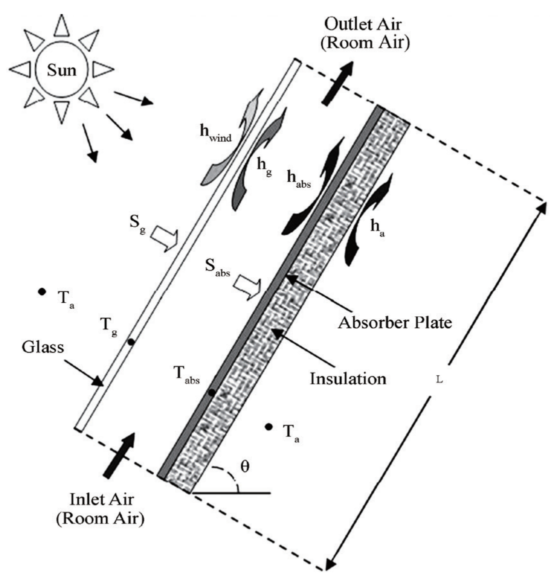

Figure 1
Solar Chimney Schematic Diagram and Heat Transfers Analysis (Maerefat & Haghighi, 2010)

The equation of energy balance for glass cover is:

$$S_g A_g + hr_{abs-g} A_{abs} (T_{abs} - T_g) = h_g A_g (T_g - T_{fsc}) + U_{g-a} A_g (T_g - T_{fsc})$$

Maerefat and Haghighi (2010) reported the overall top heat loss coefficient from the glass cover to ambient air (U_{g-a}) as:

$$U_{g-a} = h_{wind} + hr_{g-sky}$$

Duffie and Beckmann (1980) reported the convective heat transfer coefficient due to the wind as follow:

$$h_{wind} = 2.8 + 3\, u_{wind}$$

The solar radiation heat flux S_g absorbed by the glass cover is given by:

$$S_g = \alpha_g I$$

If the ambient temperature is adopted from Ong (2003): then Theradiative heat transfer coefficient from the outer glass surface to the sky is written as

$$hr_{g-sky} = \frac{\sigma\varepsilon_g(T_g + T_{sky})(T_g^2 + T_{sky}^2)(T_g - T_{sky})}{(T_g - T_a)}$$

Where the sky temperature by Duffie and Beckmann (1980) is:

$$T_{sky} = 0.0552 T_a^{1.5}$$

The radiation heat transfer coefficient between the absorber plate and the glass cover from Ong (2003) is:

$$hr_{abs-g} = \frac{\sigma\,(T_g^2 + T_{abs}^2)(T_g + T_{abs})}{\frac{1}{\varepsilon_g} + \frac{1}{\varepsilon_{abs}} - 1}$$

The convective heat transfer coefficient between the glass cover and the air flow in the chimney:

$$h_g = \frac{Nu_g k_{fsc}}{L_g}$$

Where Nusselt number is given by Mathur, Mathur, and Anupma (2006):

$$Nu_g = 0.6(Gr_g \cos\theta\, Pr_{fsc})^{0.2}$$

And Grashof number is calculated by Mathur, Mathur, and Anupma (2006) from:

$$Gr_g = \frac{g\beta S_g (L_g)^4}{k_{fsc}{v_{fsc}}^2}$$

The convective heat transfer coefficient between the inclined absorber wall and the air flow in the chimney is given by:

$$h_{abs} = \frac{Nu_{abs} k_{fsc}}{L_{sc}}$$

Where Nusselt number is given by Mathur, Mathur, and Anupma (2006):

$$Nu_{abs} = 0.6(Gr_{abs} \cos\theta\, P_{r\,fsc})^{0.2}$$

And Grashof number is calculated by Mathur, Mathur, and Anupma (2006) from:

$$Gr_{abs} = \frac{g\beta S_{abs} (L_{abs})^4}{k_{fsc}{v_{fsc}}^2}$$

Where an average surface–air temperature was used to calculate all property values related to the glass and the absorber.

The air flow in the chimney is controlled by an energy balance equation as:

$$h_{abs}A_{abs}(T_{abs} - T_{fsc}) + h_g A_g (T_g - T_{fsc}) = -\, m_a C_{fsc}(T_{fsc} - T_g)/\gamma$$

Where g is a constant and recommended as 0.74 by Maerefat and Haghighi (2010). For the absorber plate the energy balance equation is written as:

$$S_{abs}A_{abs} = h_{abs}A_{abs}(T_{abs} - T_{fsc}) + hr_{abs-g}A_{abs}(T_{abs} - T_g)$$
$$+\, U_{abs-a}A_{abs}(T_{abs} - T_a)$$

The overall heat transfer coefficient from the rear of the absorber wall to the ambient ($U_{abs\,-\,a}$) is given by Maerefat and Haghighi (2010):

$$U_{abs-a} = \frac{1}{\frac{1}{h_a} + \frac{\delta_{ins}}{k_{ins}}}$$

In the above equation h_a has been taken as 2.8 W/m².K from Duffie and Beckmann (1980). All thermo-physical properties for this part are given (Tchinda, 2003; Tchinda, 2008).

2. EVAPORATIVE COOLING CAVITY (ECC)

Researchers showed that the natural ventilation system that combined both a solar chimney and an evaporative cooling is capable of providing a temperature reduction inside a building. In addition this system is environmentally friendly and energy saving at the same time.

Figure 1 illustrates the system consisting of two parts: the solar chimney and the cooling cavity. The solar chimney based on a glass surface that is oriented to the south and an absorber wall which acts as a capturing for the solar radiation.

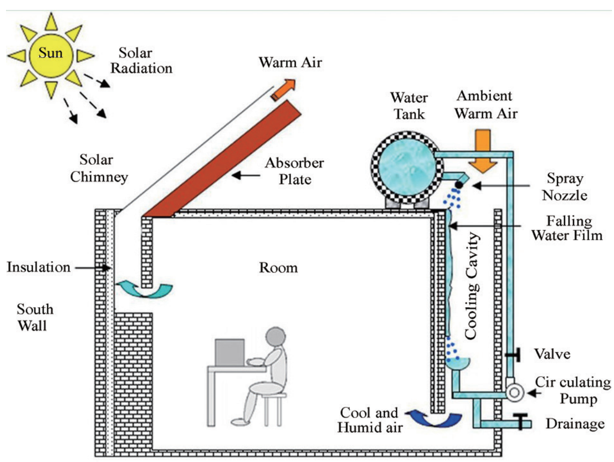

Figure 2
Schematic Diagram of Solar Chimney and Cooling Cavity (Maerefat & Haghighi, 2010)

As shown in Figure 1 (Maerefat & Haghighi, 2010), in the cooling cavity, water is sprayed into the air that enters the cavity. The water flows as a thin film along the wall surfaces of the air passage, then either recollected by a pump to the water tank or carried out by drainage. Since partial pressure of water vapor at the interface is higher than air pressure, there is a mass transfer of evaporated water into the air. This mass transfer is associated with latent heat transfer of the water vaporization. At the same time, convective heat transfer takes place due to the temperature difference between surface of the water and the air. Thus we can call it a direct evaporative passive cooling system.

Each of the solar chimney and the cooling cavity will work simultaneously as a result of combining their effects. The solar energy heats up the room air flowing through the chimney, and the hot air generates the draft in the chimney. This draft induces air ventilation in the whole system: solar chimney, room, and cooling cavity. The chimney effect causes the air to be drawn through the cooling cavity with wetted cool surfaces and to remove heat from this air and brings cooled supply air into the room.

This study only gives the acceptable temperature range of indoor air when the outdoor temperature is within the range of 40-45 °C and does not recommend the suitable rate of ventilation. However, the ventilation rate is set approximately around 3 air changes per hour to reduce possible pollution concentration and to ensure that the thermal comfort condition is provided.

2.1 Modeling the Evaporative Cooling Cavity

A simplified steady state model is developed to determine the air mean temperature at the outlet of cooling cavity and room air temperature. For modeling the cooling cavity, heat and mass transfer from the water film into the air flow and the overall energy balance equations are taken into account.

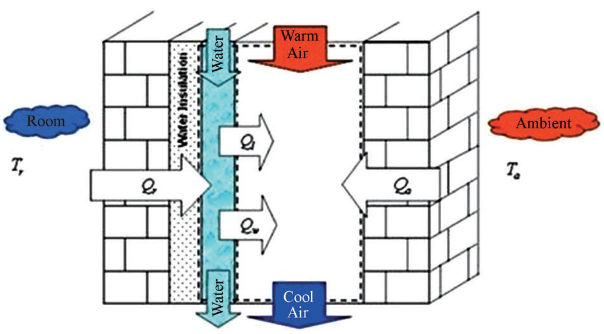

Figure 3
Schematic Diagram of Heat and Mass Transfer in the Cooling Cavity (Maerefat & Haghighi, 2010)

From the energy and mass conservation laws an additional equation is considered along with the three heat balance equations resulted from the solar chimney analysis. The convective heat transfer from the water film into the air flow is given by:

$$Q_w = h_{fcc}(T_w - T_{fcc})A$$

The following relation is adopted in this study from Maerefat and Haghighi (2010):

$$h_{fcc} = 57(u_{fcc})^{0.7}$$

The heat transfer from the room air into the water film is obtained from:

$$Q_r = U_{r-w}(T_r - T_w)A$$

The overall heat transfer coefficient from the room to the water film is calculated from:

$$U_{r-w} = \frac{1}{\frac{1}{h_r} + \frac{1}{h_w} + \frac{\delta_{wall}}{k_{wall}} + \frac{\delta_{ins}}{k_{ins}}}$$

The heat transfer from the ambient into the air flow is obtained from:

$$Q_a = U_{a-fcc}(T_a - T_{fcc})A$$

The overall heat transfer coefficient from ambient to the air flow is given by:

$$U_{a-fcc} = \frac{1}{\frac{1}{h_a} + \frac{\delta_{wall}}{k_{wall}} + \frac{1}{h_{fcc}}}$$

For steady, fully developed, two-dimensional flow of a laminar film along a vertical surface, the water side film heat transfer coefficient by Erens and Dreyer (1993) can be expressed as:

$$h_w = \frac{k_w}{\delta_w}$$

Where:

$$\delta_w = (\frac{3u_w{}^2}{4g})^{1/3}(Re_w)^{1/3}$$

$$Re_w = \frac{m_w}{b\mu_w}$$

$$u_w = 1.5\,(\frac{v_w g}{48})^{1/3}(Re_w)^{2/3}$$

Where δ_w and u_w relations are given by Maerefat and Haghighi (2010). The heat transfer from the ambient into the air flow is obtained from:

$$Q_a = U_{a-fcc}(T_a - T_{fcc})A$$

$$U_{a-fcc} = \frac{1}{\frac{1}{h_a} + \frac{\delta_{wall}}{k_{wall}} + \frac{1}{h_{fcc}}}$$

The heat transfer due to water evaporation (latent heat transfer) is given by:

$$Q_l = m_w H_w$$

$$H_w = C_v T_w + H_{fg}$$

$$H_w = C_v T_w + H_{fg}$$

The overall heat balance for the air that enters the cooling cavity equals:

$$Q_{total} = Q_w + Q_r + Q_a + Q_l = m_a C_a \left(T_a - T_r \right)$$

$$T_{fcc} = \frac{T_r + T_a}{2}$$

The relative humidity of the gas that enters the room can be calculated from:

$$\emptyset = \frac{\omega P_a}{(0.622 + \omega) P_g}$$

Where P_a is the atmospheric pressure and P_g is the gas saturation pressure at ambient temperature. The air change per hour can be expressed by Maerefat and Haghighi (2010) as:

$$ACH = \frac{3600\, m_a}{\rho_a V}$$

In actual application for this model two configurations could be done to achieve evaporative coolingas shown in Figure 4. Co-current and counter-current maybe applicable but at the same time there are some limitations in using any of them. These limitations arise from design issues related to space and cost.

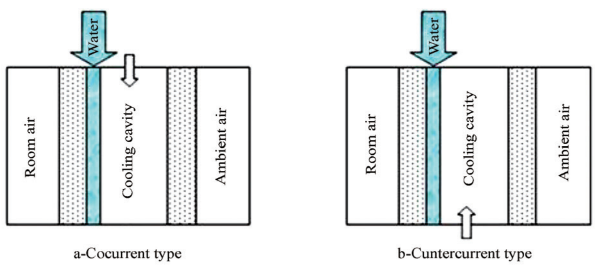

Figure 4
(a) Co-Current and (b) Counter-Current Flow (Maerefat & Haghighi, 2010)

Some postulations were assumed to enable solving the mathematical model of the SC combined with the ECC. The major assumptions are summarized as follows:

a. The air at the room is at a uniform temperature.

b. Air enters the chimney at room air temperature.

c. Only buoyancy force is considered and wind induced natural ventilation is not included.

d. The flows in the channels are laminar, and hydro dynamically and thermally are fully developed.

e. The glass cover is opaque for infrared radiation.

f. Thermal capacities of glass and wall are negligible.

g. The air flow in the channel is radioactive non-participating medium.

h. All thermo-physical properties are evaluated at an average temperature.

i. Thermal resistance of water film is negligible.

j. The spray enthalpy is negligible.

k. The air enthalpy is only expressed as a linear function of wet bulb temperature.

l. The Lewis number relating heat and mass transfer is 1.0.

m. The system is at steady-state condition.

Heat balance equation for the cooling cavity along with the three heat balance equations for the solar chimney form a system of four equations with four unknowns (T_{abs}, T_{fsc}, T_g, T_{fcc}). Using numerical solution, the value of these variables could be determined in order to find the desired temperature of the room T_r. All thermo-physical properties for this part are obtained (Tchinda, 2008; Tchinda, 2003).

3. GEOMETRY AND DESIGN SOLUTION OF THE SYSTEM

When solving the system numericallywith the flat plate solar chimney, results showed that the geometry of the solar chimney is relatively large (4m length, 1m width and 0.3m depth). So it is inconvenient and expensive to attach this solar chimney to a room like the one used in Figure 1 (4m x 4m X 3.125m).

A compound parabolic concentrator (CPC) with one-sided flat absorber is introduced to be a suitable solution for this design problem, where it was found that this new configuration can truncate the solar chimney to one third of its previous size (Tchinda, 2008).

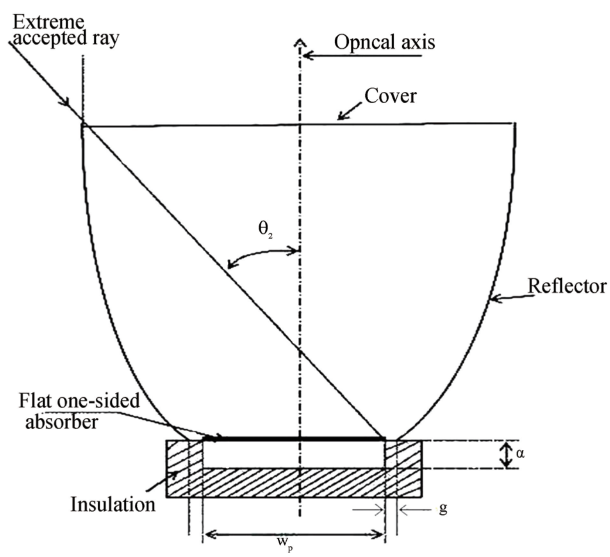

Figure 5
A Two-Dimensional CPC With a Flat One-Sided Absorber Cross Section (Tchinda, 2008)

3.1 Structure and Mathematical Modeling of the Compound Parabolic Concentrator

The CPC is capable of accepting solar radiation for long periods each day without tracking of the sun. It also has the advantage of concentrating the diffuse radiation, which is not possible using an imaging collector. The following drawing shown in Figure 5 illustrates a cross section of a CPC.

All rays within the acceptance angle (2a) fall onto the flat absorber. A certain fraction of the rays entering the aperture will reach the absorber directly, while the other rays will reach the absorber after one or more reflections from the parabolic reflectors. The ratio of the average energy flux on the receiver to that on the aperture is the concentration ratio (CR), which is given by Tchinda (2008):

$$C_r = \frac{1}{\sin\theta a} = \frac{Ac}{Ap}$$

The reflection of radiation from the parabolic reflector is taken into account by the apparent reflectance $\rho_m^{\langle n\rangle}$with $\langle n\rangle = 0.5 + (0.07\ CR)$ for a CPC with flat plate absorber (Tchinda, 2008).

The direction of the beam radiation incident on various components in the collector can be found through geometry. Any reflection from these components, particularly multi-reflections from the parabolic reflector, will cause a reorientation of rays to the effect that the ray's reflection pattern becomes exceedingly difficult to follow without reliance on a detailed ray tracing. To facilitate analysis, these reflections are treated as diffuse, and their energy is taken into account in terms of diffuse reflectivity (Tchinda, 2008). The succeeding absorption and transmission processes inside the CPC are diffusive and are taken into account in terms of the diffuse properties. The solar and infrared energy exchanges in the collector are treated separately using pertinent radiative properties in the spectrum. The physical and optical properties of materials are assumed to be independent of temperature. The following side view and an electric analogy circuit for the CPC collector is used to demonstrate the analysis.

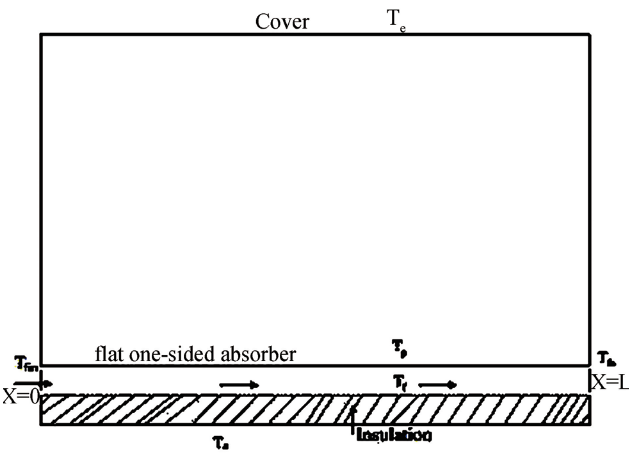

Figure 6
A Two-Dimensional CPC With a Flat One-Sided Absorber Side View (Tchinda, 2008)

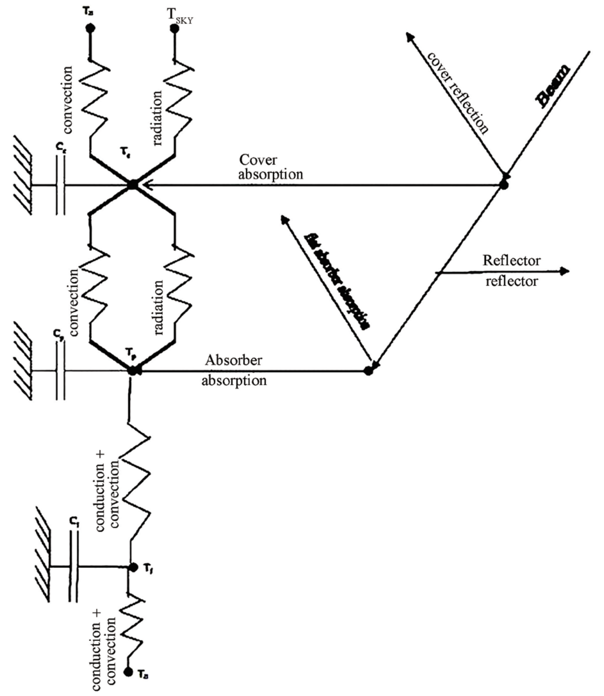

Figure 7
Electrical Analogy for Two Dimensional CPC With a Flat One-Sided Absorber (Tchinda, 2008)

Applying heat balances on the cover, the plate absorber and the fluid, the following set of partial differential equations can be derived (Tchinda, 2008):

For the cover:

$$m_c C_c \Delta T_c = q_c + hr_p(T_p - T_c) + h_{p-c}(T_p - T_c)$$
$$- hr_{sky}(T_c - T_{sky}) - h_{c-a}(T_c - T_a)$$

For the flat plate absorber:

$$m_p C_p \Delta T_p = q_p + hr_p(T_p - T_c) + h_{p-c}(T_p - T_c) - q_u)$$

For the fluid:

$$\rho_f t_f C_f \Delta T_f = q_u - \frac{mC_f}{W_p}\frac{dT_f}{dx} - U(T_f - T_b)$$

Using Hsieh's theory q_c and q_p (Tchinda, 2003) can be expressed as:

$$q_c = I[\alpha_c + \alpha_c \tau_c \rho_p \rho_m^{2<n>}]\frac{A_c}{A_p}$$

$$q_p = I\tau_c \rho_m^{<n>} P\left[\alpha_p + \alpha_p \rho_p \rho_c \frac{A_p}{A_c}\right]\frac{A_c}{A_p}$$

Where P is the gap loss factor which is equal to (1-g/W_p), and g here is the gap thickness. Also A_c = W*L and A_p = W_p*L.

$$q_u = U_f(T_p - T_f)$$

$$U_f = \frac{N_{uf} k_f}{D_h}$$

Calculations were made using the previous equations and the properties (Tchinda, 2003).

4. RESULTS AND DISCUSSION

Results were obtained by applying energy and mass balance on a system consisting of a flat plat solar chimney combined with an evaporative cooling cavity. Also taking in consideration the volume of a room located in Amman, Jordan. Using all these conditions in addition with the properties of the fluids and the materials forming the two systems (Tchinda, 2003), numerical codewas constructed to solve for the temperature of the room reached in steady state condition. As aresult the temperature was dropped to 25.6 ˚C, which satisfies the goal of cooling and ventilating the room to the comfort conditions needed. But on the other hand, these results were based on a solar chimney of length (4m), width (1m), and depth of (0.3m), which is relatively large. As it mentioned before the solar chimney was modified with a compound parabolic concentrator (CPC) in order to minimize the dimensions of the solar chimney without changing the value of the air mass flow rate. Studying the design parameters of the CPC (Tchinda, 2008) and keeping in mind the need to have the same air mass flow rate, and applying the energy and mass on this configuration resulted in the following:

- First: with a solar radiation (I) equals 500 W/m^2, heat absorbed by the cover (q_c) was 120 Watt and heat absorbed by the absorber (q_p) was 1439 Watt.
- Second: with a chimney inlet air flow temperature (T_{fi}) equals 30˚C the outlet temperature ($T_{f,o}$) was found to be 50˚C.
- Third: the temperature of the flat plate absorber (T_p) equipped with the CPC reached 128˚C.

These results, considered to be theoretical, have matched the results of a previous experimental study done on the thermal behavior of a solar air heater with compound parabolic concentrator (Tchinda, 2008). The behavior of the system is demonstrated in the next graphical relations (Figures 8 and 9).

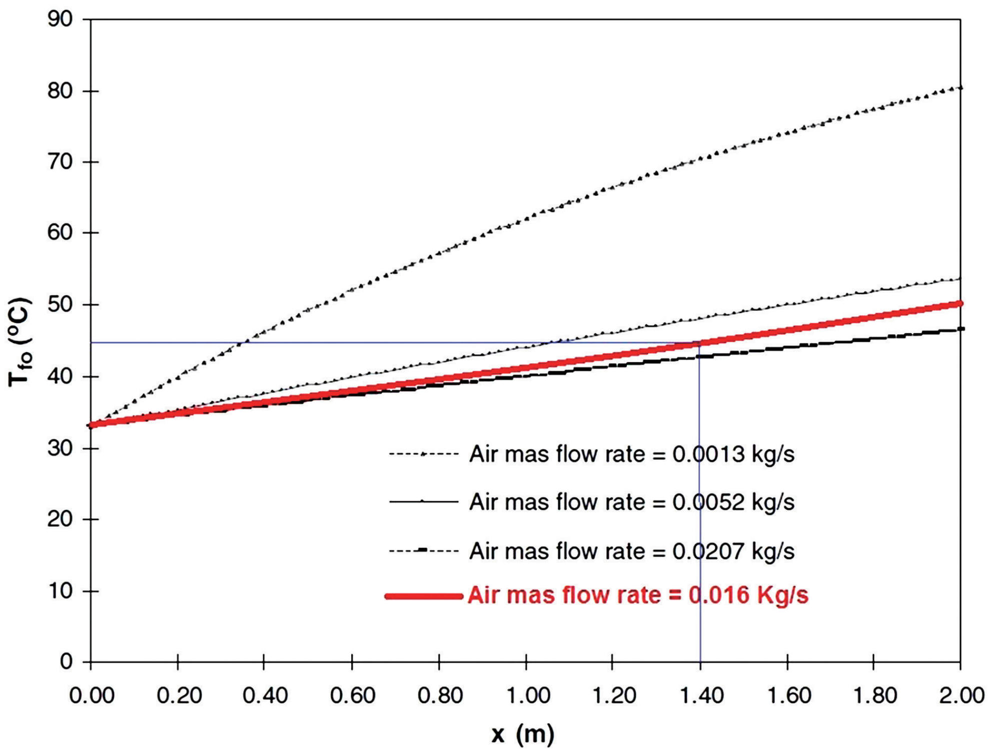

Figure 8
Effect of the Air Mass Flow Rate on the Local Temperature in the Flow Direction at L = 1.4 m and T_{fi} = 33˚C

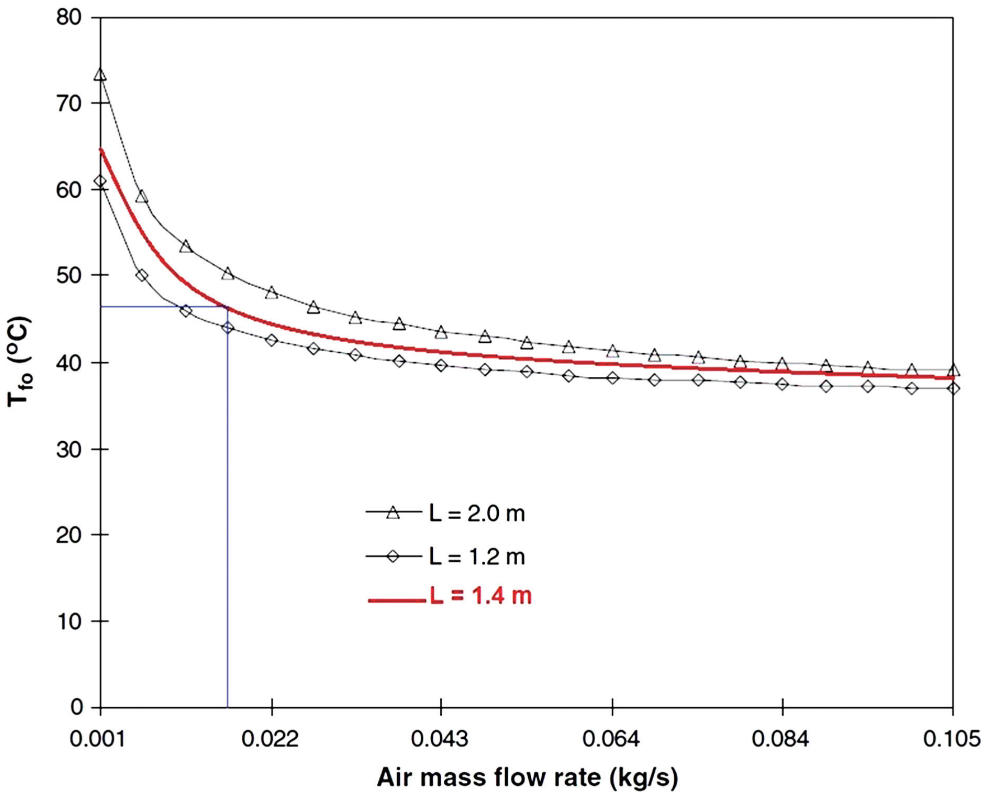

Figure 9
The Outlet Air Temperature as a Function of the Mass Flow Rate for Different Values of L

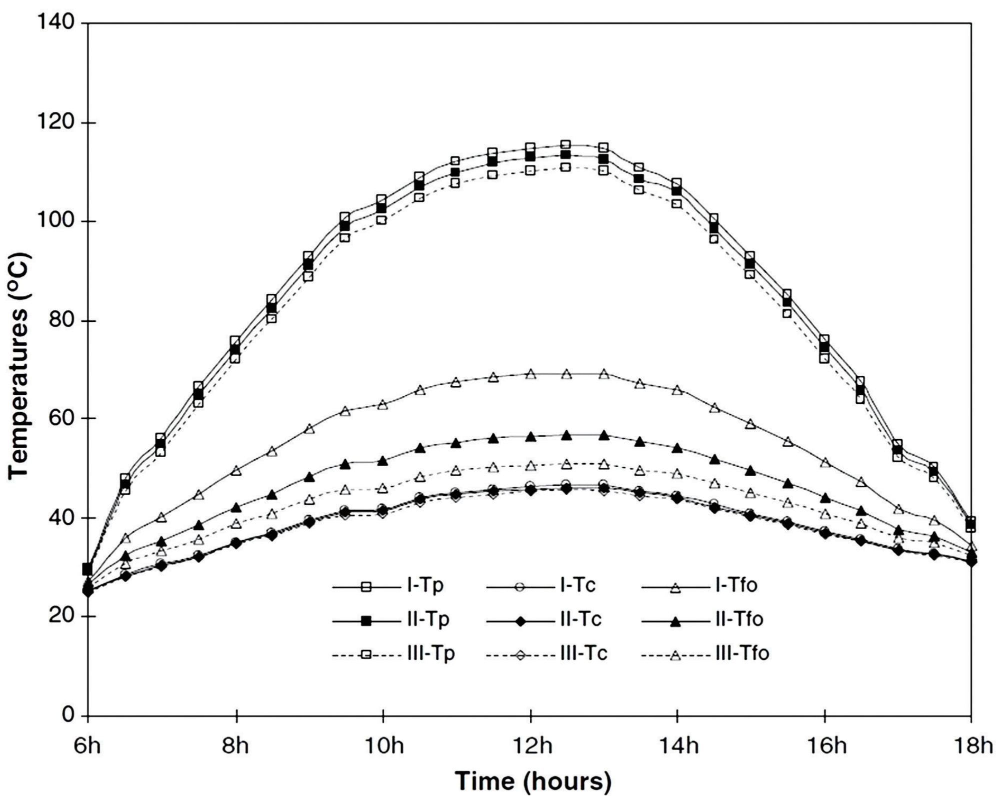

Figure 10
Effect of the Air Mass Flow Rate on the Hourly Variations of Temperatures of the Collector (I) Air Mass Flow Rate = 0.0013 kg/s (II) Air Mass Flow Rate = 0.0065 kg/s (III) Air Mass Flow Rate = 0.013 kg/s all at L = 2.0 m

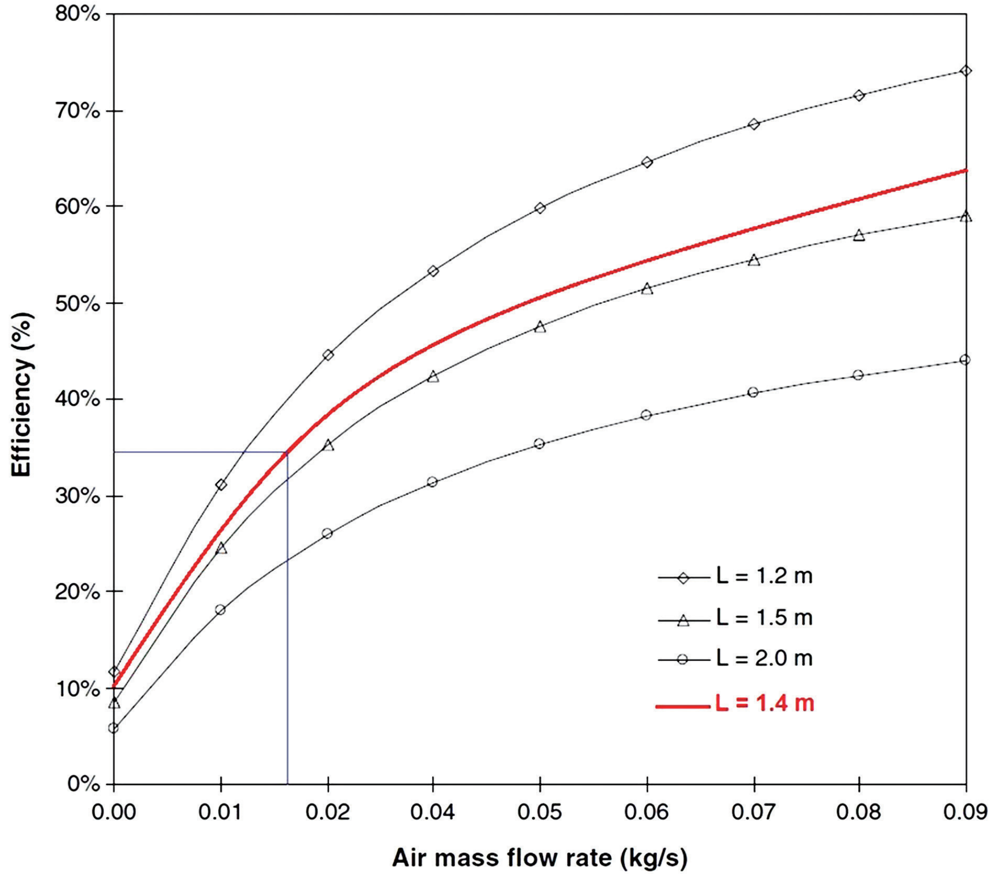

Figure 11
Effect of the Air Mass Flow Rate on the Daily Efficiency of the Collector for Different Values of Collector Length L

The dimensions of the CPC and the parabola equation that forms the reflector used in this study shown in the next sketch:

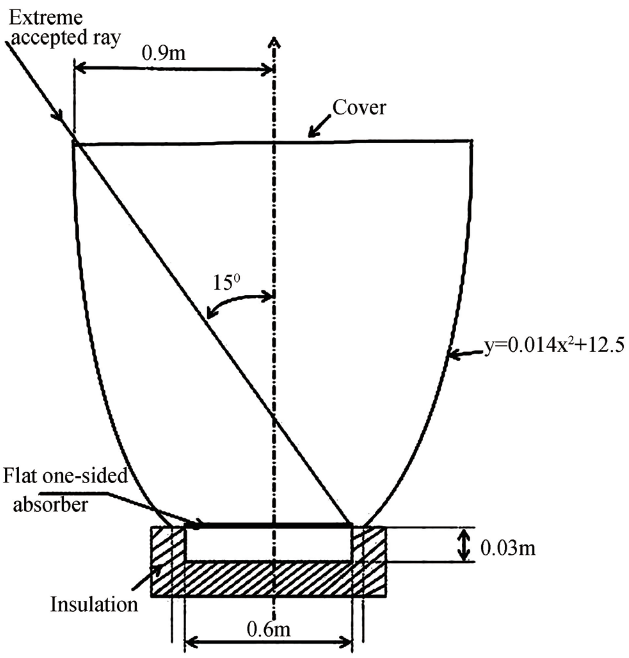

Figure 12
Dimensions of the CPC Used in This Analysis (Tchinda, 2008)

The efficiency of a solar chimney or a CPC can be found through this equation by Tchinda (2008):

$$\eta = \frac{Q_u}{AI}$$

Where

$$A_{sc} = L_{SC} \times W_{SC}$$

$$A_{CPC} = A_C = L_C \times W_C$$

Efficiency of the flat plate solar chimney is found to be around (8%), on the other hand the efficiency of the CPC was about four times higher (33%) since A_{SC} is (63%) larger than A_{CPC} and $Q_u$of the flat plate solar chimney is (33%) lower than $Q_u$of the CPC.

Regarding the evaporative cooling cavity, an ambient air with temperature, relative humidity and wind velocity equals to 34 °C, 25% and 4.34m/s, respectively, is used in this system. Since the cavity depends on evaporative cooling, the process can be presented by the following figure:

The process in the previous psychometric chart represents the evaporative cooling which goes in a straight lines parallel to the wet bulb or saturation temperature

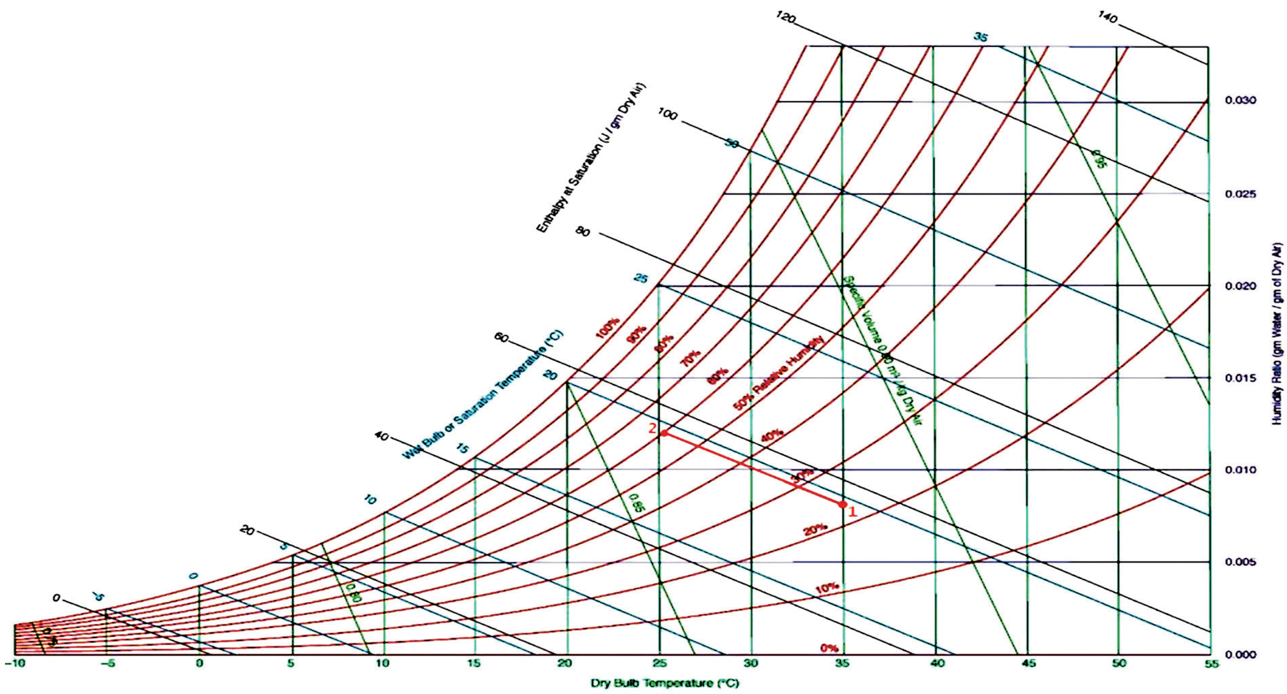

Figure 13
Psychometric Chart Representing Air Cooling and Humidification in the Cavity

lines as a result of the latent heat transfer. State 1 at 35°C and 25% relative humidity represent the ambient air and state 2 at 25°C and 60% relative humidity represent air after being cooled by the cooling cavity. As it shown, a temperature difference of 10°C is created by the evaporative cooling method. Also a final relative humidity of 60% is achieved by spraying water into the cavity. A condition of such temperature and relative humidity can insure human comfort which is the main purpose of this study. Numbers represented in this chart has gone along with the results obtained by the numerical code

CONCLUSION

Integrating ventilation and cooling system with a solar chimney and an evaporative cooling cavity is found compatible for dried climate areas to provide the comfort condition for humans.

Using a solar chimney as a driving force for the system demands that it generates a certain air mass flow rate. The magnitude of this air mass flow rate depends on the cooling load, dimensions of the room, targeted room temperature and the length of the solar chimney.

Increasing solar chimney length more than the needed for the required air mass flow rate results in discomfort and more cost because extra length can't afford more heating.

The compound parabolic concentrator is capable of providing the same performance related to temperature and flow rate with reduction in overall dimensions, which make it a better choice rather than the flat plate solar collector.

Regarding the evaporative cooling cavity, it is not necessary to use recirculating pump in order to compensate the loss in water used to cool air since the mass flow rate that achieves this cooling is relatively small and approximated about 25 liter per day. Such amount could be refilled from time to time.

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NOMENCLATURE

A: area (m²)
ACH: air change per hour (h^{-1})
b: width of cooling cavity (m)
C: specific heat (J/kg.K)
c: pressure loss coefficient of fittings
D: gap depth between absorber wall and glass (m)
d: diameter (m)
H: enthalpy (kJ/kg)
h: convective heat transfer coefficient (W/m².K)
hr: radiative heat transfer coefficient (W/m².K)
I: total incident solar radiation on south facing inclined surface (W/m²)
k: thermal conductivity (W/m.K)
L: length (m)
m: mass flow rate of air (kg/s)
P: pressure (Pa)
Q: heat transfer to air stream (W/m²)
R: thermal resistance (m².K/W)
r: radius (m)
S: solar radiation heat flux absorbed by plate or glass (W/m²)
T: temperature (K)
t: thickness (m)
U: overall heat transfer coefficient (W/m².K)
u: air velocity (m/s)
V: volume of room (m³)
W: width (m)
x, y: coordinate system (m)
Z: height of chimney inlet (m)

GREEK SYMBOLS

α: absorption coefficient
β: volumetric coefficient of expansion (K^{-1})
γ: constant equal to 0.74
δ: heat penetration depth (m)
ε: emissivity
η: efficiency
θ: angle of inclination
λ: thermal diffusivity (m²/s)
μ: Dynamic viscosity (kg/s.m)
ν: Kinematic viscosity (m²/s)
ξ: friction factor
Ø: relative humidity (%)
ρ: density (kg/m³), reflectance
τ: transmissivity
σ: SteffaneBoltzmann constant (5.67×10^{-8} $W/m^2.K^4$)
ω: frequency of temperature oscillation (rad/s)

DIMENSIONLESS TERMS

Gr: Grashof number [$g\ \beta_f\ (T - T_f)\ L^3/\nu^2$]
Nu: Nusselt number [$h_f\ L/\mu_f$]
Pr: Prandtl number [$C_f\mu_f\ /k_f$]
Ra: Rayleigh number [GrPr]
Re: Reynolds number [$u_f\ D_h/\nu_f$]

SUBSCRIPTS

a: ambient
abs: chimney absorber wall
c: convective, cover
cc: cooling cavity
f: air flow
fg: latent heat
g: glass
h: hydraulic
i: internal
in: inlet
ins: insulation
l: latent
m: mirror
o: outlet
p: CPC absorber wall
r: radius, room
s: soil
sc: solar chimney
st: inner surface of tube
su: undisturbed soil
t: pipe
u: useful
v: vapor
w: water