ORIGINAL RESEARCH

Integration of evaporative cooling technique with solar chimney to improve indoor thermal environment in the New Assiut City, Egypt

Amr Sayed Hassan Abdallah[1,2]*, Hiroshi Yoshino[2], Tomonobu Goto[2], Napoleon Enteria[2], Magdy M Radwan[1] and M Abdelsamei Eid[1]

Abstract

Cooling buildings in summer is one of the main environmental problems for architects and occupants in many hot dry countries. The summer temperature during these countries reaches peaks of more than 40°C in some. Mechanical air conditioners can solve the problem, but they put a heavy strain on the electricity consumption. Egypt in general has rich sunny and clear skies. Therefore, these conditions encourage to enhance evaporating with natural ventilation and save energy. This paper develops an integration of direct evaporative cooling tower with a solar chimney multi-zone thermal ventilation model. Simulation is done using commercial couple multi-zone airflow under COMIS-TRNSYS software (Madison, WI, USA) to assess natural ventilation and indoor thermal comfort. The results show that the system generates 130.5 m^3/h under the effect of solar radiation only and minimum 2 ACH without pressure coefficient which is considered the minimum requirement of ACH. The findings show that the new integrated system interacts with the building envelope and weather conditions to achieve a decrease in indoor temperatures that reach 10°C to 11.5°C compared to outdoor temperatures.

Keywords: Passive cooling; Solar chimney; Integration system; TRNSYS-COMIS; Multi-zone models

Background

Buildings contribute over 40% of the total global primary energy use corresponding to 24% of the world CO_2. Heating, ventilation, and air conditioning (HVAC) system is the technology of indoor and vehicular environmental comfort which is responsible for about half of the energy use in buildings [1]. The quest to reduce environmental impacts of conventional energy resources and, more importantly, to meet the growing energy demand of the global population had motivated considerable research attention in a wide range of environmental and engineering application [2]. Effective integration of passive features into the building design can significantly minimize the air conditioning demand while maintaining thermal comfort [3]. Passive evaporative cooling is one of the most efficient and long recognized ways of inducing thermal comfort in predominantly hot dry climates. Historically, evaporative cooling was used extensively in traditional architecture throughout the world's hot arid countries [4-6]. In order to extend the use of evaporative cooling, short wind tower cooling was integrated with inclined solar chimney to increase the pressure difference and the system efficiency. This was done after observing the indoor discomfort for one of the Egyptian houses built in the New Assiut City [7].

Many researches have been conducted on using natural ventilation and evaporative cooling strategies for producing cool air and the effect of using a solar chimney on thermal-induced ventilation in buildings. Three important researches were under taken to study these passive systems. Bahadori introduced the idea of capturing the wind within a tower and then pass it through wetted conduit walls [4]. Maerefat and Haghighi also put forward a new solar system employing a solar chimney together with an evaporative cooling cavity. The numerical calculation showed that this integrated system with

* Correspondence: arch_amrsayed@yahoo.com
[1]Department of Architecture, Assiut University, P.O. Box 71516, Assyut, Egypt
[2]Graduate School of Engineering, Tohoku University, Aoba 6-6-11-1201, Sendai 980-8579, Japan

the proper configurations was capable of providing good indoor conditions during daytime in the living room even at a poor solar intensity of 200 W/m^2 and a high ambient air temperature of 40°C [8]. Alemu et al. developed an integrated model incorporating passive airflow components into a coupled multi-zone ventilation and building thermal model. This model allows an assessment of a combination of passive features such as solar chimney and wind-induced earth-air tunnel for both natural and hybrid ventilation systems at the design stage [9]. A lot of awareness is ongoing worldwide on solar energy utilization [10].

The main objective of this paper is to develop an integrated solar chimney with passive cooling strategies into a coupled multi-zone ventilation and building thermal model. This was done by developing a mathematical model to incorporate passive features into a multi-zone ventilation model incorporating a simple thermal model using COMIS-TRNSYS software (Madison, WI, USA). This development system is used in a room of a single zone to study the performances and the advantages of an integrated system in the numerical model. Also, it focuses on the simulated performances and its impact on the comfort of occupants for the summer period with an integrated proposed system.

A theoretical study of the direct evaporative wet medium

An evaporative cooler (EC) is a device that cools air by passing it across a wet medium. As hot air passes over the wet medium, some of its energy goes to evaporate water from the medium. The air comes out of the other side of the device having given up some of its energy and consequently exits at a lower temperature with higher humidity and constant enthalpy. The minimum temperature that can be reached is the thermodynamic wet bulb temperature (TWBT) of the incoming air. It was reported that half of the conventional cooling device in the south west of the USA was replaced by simple direct evaporative air coolers; this would save 18 million barrels of oil per year [5,11].

In this paper, heat and mass transfer from the wet medium into the airflow are taken into account. Based on the theoretical study and initial system conditions, evaporative cooler performance is based on the concept of the adiabatic process. This means no heat is added to or removed from the system; the process of exchanging the sensible heat of the air for latent heat of evaporation from the water is an adiabatic process [12].

A description of the system

The performance of the system is to provide desired comfortable conditions and a suitable rate of ventilation, depending on several parameters such as the ambient conditions (temperature, solar radiation, relative humidity, wind speed, and pressure coefficient). The following dimensions and specifications are applied to the model: The system is located in the New Assiut City, Egypt. New Assiut City is located at 27.30°N latitude and 31.15°E longitude. The solar chimney is oriented towards the south. The calculations were carried out for a single zone and to study the integration of the new proposed system, having a dimension of 4.0 m × 4.0 m × 3.125 m. This system was developed from the reference system of inclined solar chimney attached to a room with an evaporative cooling cavity (ECC) [8]. The old model assumes zero pressure boundaries at the inlet and outlet except for the stack effect. Therefore, it is not taken under consideration the variety of the buildings' static pressure from the atmospheric pressure and the effect of wind pressure coefficient on buildings. Also, there are many reasons that make it very difficult to integrate this system into Egyptian housing. The ECC strategies consumed a lot of water because of water film and a lot of space. Figure 1 shows the diagram of the reference system and the new proposed system.

Nevertheless with the new proposed system, the inclination length of the chimney is 2 m on the inclined angle in order not to extend the vertical height (1.5 m) according to Egyptian Building Regulation Law, and the height of the cooling tower is 1 m. The chimney air gap is 0.2 m, and the inclination of the chimney is 50° based on the angle of the reference system. Table 1 shows the parameters of the solar chimney. Figure 2 shows the location of the room and its dimension in relation to a northern orientation. The performance of the system was monitored at a steady state of condition and then 5 days in the month of June (19, 20, 21, and 23) were selected for studying the performance of the solar chimney between sunrise and sunset in order to study the timewise to focus on the hottest day in the summer season (20 June). Then, parametric studies for solar chimney and wind tower are needed in the future studies in order to choose the best dimensions for a new proposed compact design. Some parameters used in the calculations of solar chimney appear in Table 1.

The evaporative pad provides a large water surface, and the pad is wetted by dripping water from the above source. The EC works by a concentric float valve, which opens when the water level is low in the collecting grill, allowing more water to enter. When the water level returns to the full level, the valve is shut automatically.

Methods

The system is being developed with a solar chimney and a small evaporative cooling wind tower. The experimentation would become time consuming and very expensive. Therefore, the numerical model is made and developed for the system. The developed model is implemented in
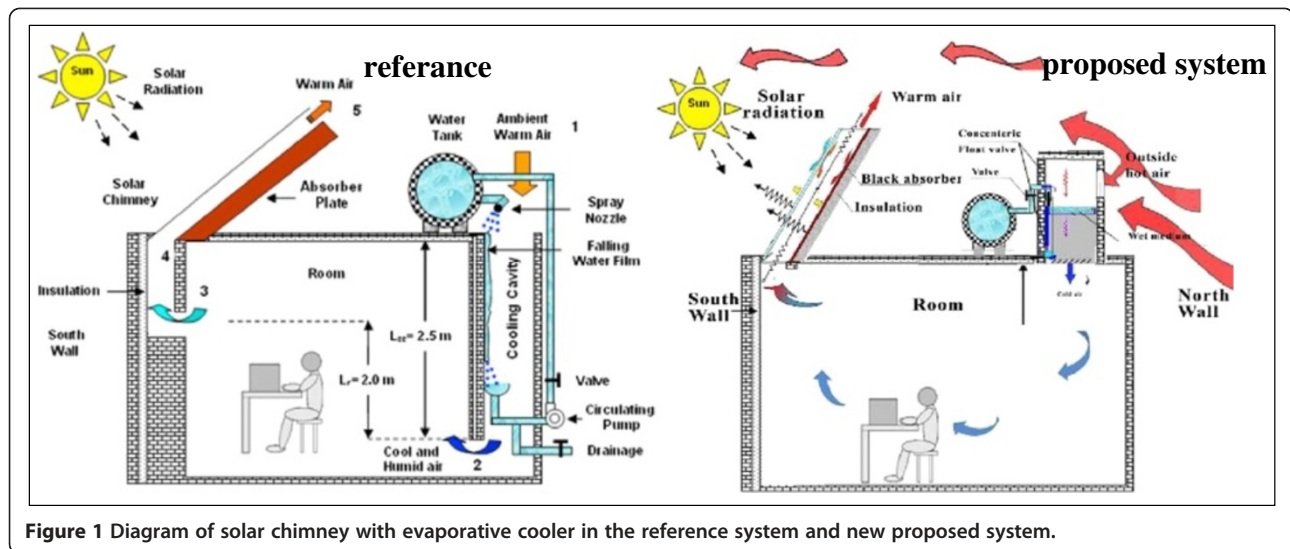
Figure 1 Diagram of solar chimney with evaporative cooler in the reference system and new proposed system.

the COMIS-TRNSYS simulation software. This direct evaporative cooling tower used a component 506d-TESS library which uses a wet medium at the top, and this component was assembled and validated by the Thermal Energy System Specialists, LLC (Madison, WI, USA) and was modified in 2004. The present model includes special airflow component which needs simultaneous prediction of temperature and airflow rates. In the developed multizone ventilation model, a building is idealized as a system of zones, openings, and ducts linked together by discrete airflow paths. Zones are represented by nodes. Chimney and evaporative cooling channels are represented by ducts for resistance and pressure drop calculations. A hydrostatic condition is assumed in zones, and the flow rate in each link is defined as a function of zone pressure, which results in a system of nonlinear equation solver, defined by the mass conservation for each zone [13]. Figure 3 shows the flow chart solving procedure of TRNSYS-COMIS.

The equations are solved numerically to predict zone pressures, and zone pressures are back substituted in the flow equations to predict the airflow rate at each link [13]. The detailed equations used for large openings and ducts are presented with the fundamentals of the multizone airflow model (COMIS, [14,15]. A description of the mathematical models used in each component can be found in the TRNSYS 16 manual handbook. Multi-

zone airflow network models deal with the complexity of flows in a building by recognizing the effects of internal flow restrictions. They require extensive information about flow characteristics and pressure distributions [16]. The pressures are calculated in the previous steps as an initial guess for pressures in the next step. Initial zone densities are based on the pressures found in the last step. The solver updates zone densities as it updates zone pressures [13]. Figure 4 shows the flow chart solving procedure for COMIS-TRNSYS.

Pressure calculation and mass flow rate through openings
To estimate the amount of air flow through an opening, it is necessary to know the pressure difference across the opening and its effective flow area. The pressure at an opening can be due to wind, as well as buoyancy, and it is therefore determined by the location of the opening, as well as the internal and external environmental parameters such as wind velocity, pressure coefficient, and outdoor temperatures.

As solar radiation passes through the glazing and absorbed at the wall surface. The air in the chimney is then heated by convection and radiation from the absorber. The decrease in density experienced by the air causes it to rise, whereupon it is then replaced by air from below, i.e., from the attached room. The rate at which air is drawn through the room depends upon the buoyancy force experienced (i.e., depending upon the temperature differential), the resistance of flow through the chimney, and the resistance to the entry of fresh air from the room. While in a wind tower, the opening faces north towards the preferable wind. This compact design with dimensions of 1 m (width) × 0.7 m (depth) and a height of 1 m depends on solar radiation and solar chimney that sucks air from the room and increases the air flow inside the room.

Table 1 The parameters used in the calculation [7]

Parameters	Value (dimensionless)
Transmissivity of glass	0.84
Absorptivity of glass	0.06
Emissivity of the glass	0.90
Absorptivity of absorber wall	0.95
Emissivity of the absorber surface	0.95
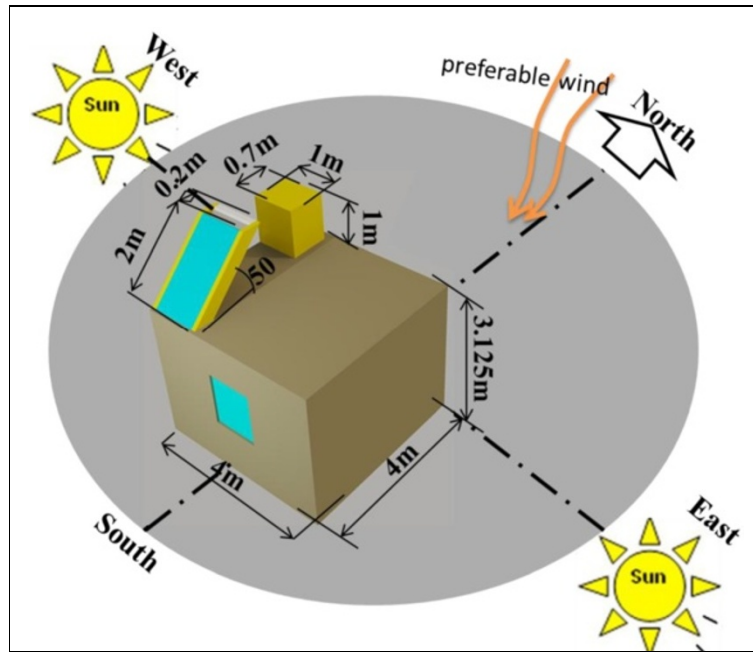
Figure 2 The location of the room with the integrated system according to north orientation.

The ventilation rate inside the system is calculated according to six variables (P_2, P_3, P_4, P_5, P_6, and P_7) which are solved from Equations 1 to 6, taking into account the base pressure on the ground ($P = 0$) as presented in Figure 3.
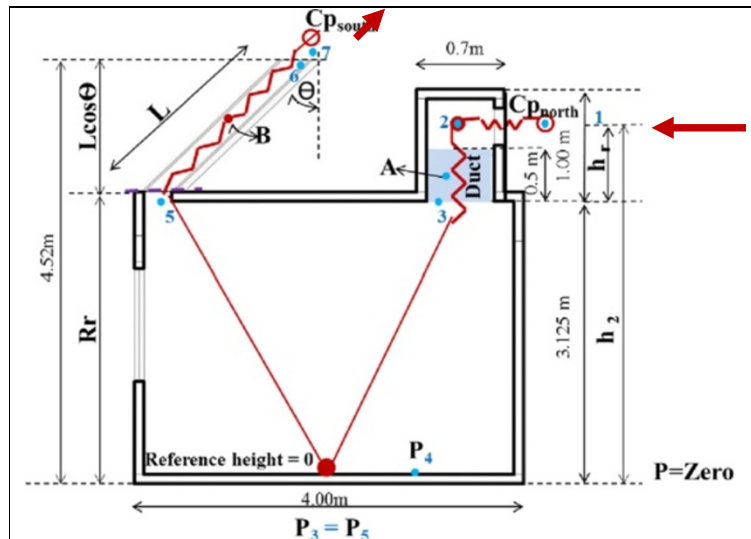
$$P_2 = \left(-\rho g h_2 + 0.5\rho_1 C p_{\text{north}} v^2\right) - \frac{\rho_2}{2}\left(\frac{Q}{\alpha_1 A_1}\right)^2, \qquad (1)$$

where $P_1 = \left(-\rho g h_2 + 0.5\rho_1 C p_{\text{north}} v^2\right)$ is the pressure at point 1 N/m^2, α is the discharge coefficient of the

opening in the wind tower, and A_1 is the area of the opening (1 m^2).
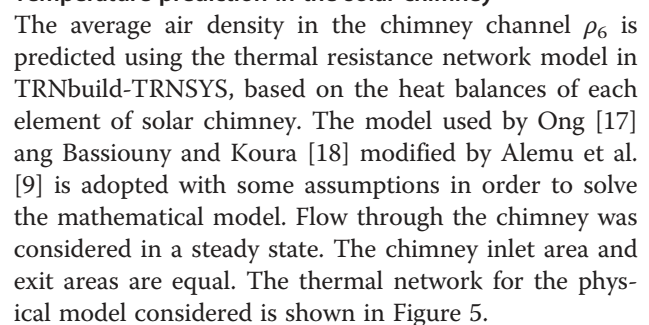
$$P_3 = P_2 - (\zeta_{\text{WetMediumInlet}} + \zeta_{\text{DuctOutlet}} + \zeta_{\text{DuctFriction}})\frac{\rho_3}{2}v_A^2$$
$$+ \rho_A g h_r, \qquad (2)$$

where the local pressure losses in the inlet $\zeta_{WetMediumInlet}$ of the evaporative cooling (wet medium), the friction of

Figure 3 A zone connected to a roof solar chimney with wind tower.

Figure 4 Flow chart solving procedure of TRNSYS-COMIS.

duct $\zeta_{DuctFriction}$, and local losses of the duct outlet $\zeta_{DuctOutlet}$ can be calculated according to the dimensions of dynamic losses due to duct fitting, Re, roughness, duct dimension, and λ. It can be determined in details from COMIS [13]. Also, the indoor velocity equals $V_A = \frac{Q}{A}$ (m/s).

$$P_4 = P_3 - \rho_3 g R_r \tag{3}$$

$$P_5 = P_4 + \rho_4 g R_r \tag{4}$$

$$P_6 = P_5 - \left(\zeta_{\text{ChimneyInlet}} + \zeta_{\text{ChimneyOutlet}} + \zeta_{\text{ChimneyFriction}} \right) \frac{\rho_B}{2} v_B^2$$
$$+ \rho_B g (L \cos\theta), \tag{5}$$

where the local pressure losses $\zeta_{ChimneyInlet}$ in the inlet of a solar chimney, the friction of chimney $\zeta_{ChimneyFriction}$, and local losses of the chimney outlet $\zeta_{ChimneyOutlet}$ can be calculated in a similar approach as used in evaporative cooling. The solar chimney is treated in the COMIS program as a rectangular duct connects to the room from one side and a node with volume that equals to zero from the other side to transfer the calculated temperature of the

chimney from TRNSYS and then another rectangular duct to outside.

$$P_7 = P_6 - \frac{\rho_6}{2} \left(\frac{Q}{\alpha_6 A_6} \right)^2, \tag{6}$$

where $P7 = -\rho g (R_r L \cos \theta) + 0.5 \rho_1 C p_{\text{south}} v^2$, and $\rho = \frac{353.25}{T} = \frac{353.25}{\theta + 273.15}$ is the air density (kg/m^3). Then, the pressure differences across the opening due to buoyancy and wind are required and solved with each others in order to predict the pressure of the room and ventilation rate Q (m^3/s).

Temperature prediction in the solar chimney

The average air density in the chimney channel ρ_6 is predicted using the thermal resistance network model in TRNbuild-TRNSYS, based on the heat balances of each element of solar chimney. The model used by Ong [17] ang Bassiouny and Koura [18] modified by Alemu et al. [9] is adopted with some assumptions in order to solve the mathematical model. Flow through the chimney was considered in a steady state. The chimney inlet area and exit areas are equal. The thermal network for the physical model considered is shown in Figure 5.

The heat balance equations from the thermal network at each element of the solar chimney are described in Equations 7, 8, and 9:

$$T_g(\text{glass cover}): \quad S_g + h_{\text{rwg}}(T_w - T_g) + h_g(T_f - T_g)$$
$$= U_g(T_g - T_a) \tag{7}$$

$$T_{f(\text{Air})}: \quad h_w(T_w - T_f) = h_g(T_f - T_g) + q'' \tag{8}$$

$$T_w(\text{absorber}): \quad S_w = h_w(T_w - T_f) + h_{\text{rwg}}(T_w - T_g)$$
$$+ U_w(T_w - T_a). \tag{9}$$

The energy balance on the air flowing through the chimney q'' with a length L and width of the air channel:

$$q^{\text{WL}''} = \dot{m} \, C_f(T_{\text{airout}} - T_{\text{airin}}), \tag{10}$$

where $\dot{m}$ is the mass flow rate of the air component. The mean air temperature T_f is given by

$$T_{f(\text{air})} = \gamma T_{\text{airout}} + (1 - \gamma) T_{\text{airin}}, \tag{11}$$

where γ (mean temperature weighting factor) = 0.75 [19].

Consider the air inlet to the chimney that has a temperature equal to the room average temperature, substitute in the above equation of q''_{WL}:

$$q''_{\text{WL}} = \dot{m} \, C_f \frac{T_{f(\text{air})} - T_{\text{airin}}}{\gamma}. \tag{12}$$
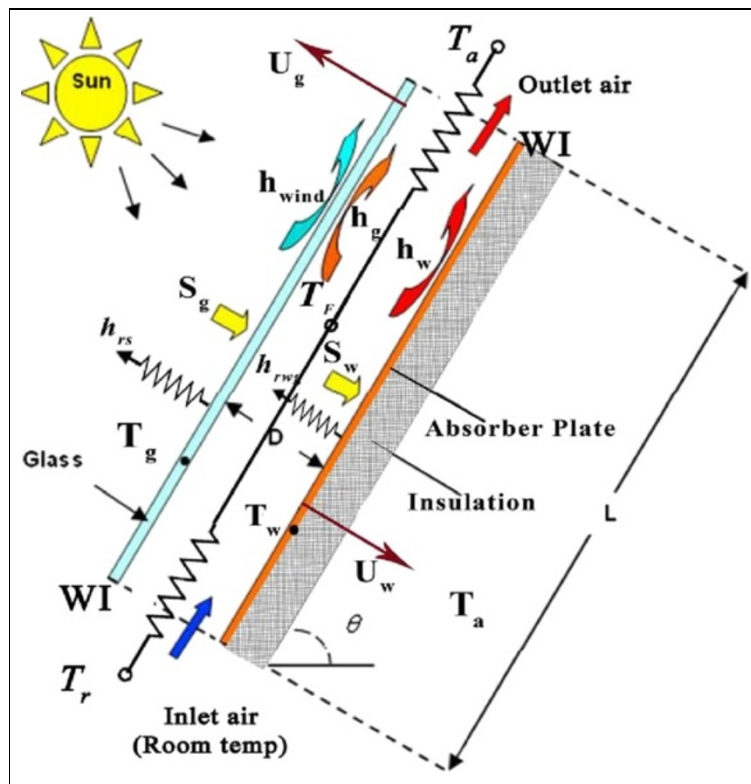
The heat transfer coefficient from the glass to atmospheric air

$$U_g = h_{\text{rs}} + h_{\text{wind}}. \tag{13}$$

The radiation heat transfer coefficient between the glass and the sky is

$$h_{\text{rs}} \frac{\sigma_g \varepsilon_g g (T_g + T_{\text{sky}}) (T_g^2 + T_{\text{sky}}^2)(T_g - T_{\text{sky}})}{T_g - T}, \tag{14}$$

where σ = solar absorptance, ε = emissivity, g = gravitational acceleration (9.81), $T_{\text{sky}} = 0.0552 \, T_a^{1.5}$.

Figure 5 Thermal network for solar chimney with air flow.

The convective heat transfer outside the glass due to the wind is

$$h_{\text{wind}} = 5.7 + 3.8 V_{\text{(wind speed at reference height)}}. \tag{15}$$

The radiation heat transfer coefficient between wall and glass h_{rwg} is

$$h_{\text{rwg}} = \frac{\sigma \left(T_{\text{g}} + T_{\text{w}}\right)\left(T_{\text{g}}^2 + T_{\text{w}}^2\right)}{1/\varepsilon_{\text{g}} + 1/\varepsilon_{\text{w}} - 1}. \tag{16}$$

The heat transfer coefficient between the absorber and outside air is

$$U_{\text{w}} = \frac{1}{1/h_{\text{w}} + \Delta\omega_{\text{insulation}}/k_{\text{insulation}}}, \tag{17}$$

Where Δw is the thickness of insulation, $k_{\text{insulation}}$ is the thermal conductivity

The convective heat transfer coefficient between air and absorber plate or glass ($h_{\text{air.absorber}}$, $h_{\text{air.glass}}$)

$$h_{\text{w.x}} = \frac{Nu_{\text{air.x}} k_{\text{air}}}{d_{\text{chimney.gap}}}, \tag{18}$$

where x refers to absorber plate or glass, and d is the channel gap.

The Nusselt number correlation for a constant heat flux on one side of the channel [20] is

$$Nu_{\text{air.x}} = 0.9282 \, Ra_{\text{air.X}}^{0.2035} \left(\frac{d_{\text{ChimneyGap}}}{Dl}\right)^{0.8972}, \tag{19}$$

where $Ra_{\text{air.x}}$ is the Rayleigh number for constant heat flux, Dl is the length.

The solar energy absorbed by the glass is

$$S_{\text{g}} = \sigma_{\text{g}} I_{\text{incident radiation}}. \tag{20}$$

The solar energy absorbed by the absorber

$$S_{\text{w}} = \sigma_{\text{w}} \tau_{\text{g}} I_{\text{incident radiation}}. \tag{21}$$

substituted the above equations into Equations 7, 8, and 9 and the three equations solved simultaneously to give the $T_{\text{g(glass cover)}}$, $T_{\text{f(Air)}}$, $T_{\text{w(absorber)}}$.

The mean air temperature $T_{\text{(Air)}}$ in the solar chimney channel is used to determine the average air density ρ_B which is then substituted in Equation 5; then, the flow rate in the solar chimney is a function of the chimney temperature.

Temperature predictions in an evaporative cooling wind tower

An evaporative cooling wind tower (ECWT) is used to let the hot dry air to enter from the outside and is cooled when it passes over a wet medium. When the air passes through the evaporative cooler, the component

calls the TRNSYS psychrometrics to fully determine the state of the inlet air. The psychrometrics returns the inlet air wet bulb temperature and enthalpy backs the inlet temperature, pressure, relative humidity, and humidity ratio. The performance of the evaporative cooler is based on the concept of adiabatic process. The wet bulb temperature of the air entering and exiting the device is assumed to be constant, where the minimum outlet temperature equals the inlet wet bulb temperature. Figure 6 shows evaporative cooler schematic.

The energy balance equation expressing the heat removed from the air to evaporate the water is expressed as

$$m_{1a} h_{1 \text{ enthalpy.AirIn}} = m_{2a} h_{2 \text{ enthalpy.Airout}} + m_{2\text{vapour}} h_{2\text{vapour}}. \tag{22}$$

The rate of heat transfer from air to water in the wet panel surface, Q_{sensible} (the heat exchange by temperature difference), is given by Δy

$$Q_{\text{s}} = h \, A \, \Delta T_{\text{LM}} = m c_{\text{p}} (T_{\text{r}} - T_{\text{a}}), \tag{23}$$

and the heat gain of latent heat of air (heat exchange by humidity ratio) can be expressed as

$$Q_{\text{L}} = m_{\text{L}} h_{\text{L}} (\Delta w), \tag{24}$$

where w is the humidity ratio.

ΔT_{LM} is the log mean temperature difference which is given by [12,21]

$$\Delta T_{\text{LM}} = \frac{(T_{\text{r}} - T_{\text{a}})}{\ln(T_{\text{r}} - T_{\text{wb}})/(T_{\text{a}} - T_{\text{wb}})}. \tag{25}$$

The performance of the systems with different flow type can be assessed on the saturation efficiency as

$$\eta \, \text{Saturation} = \frac{T_{\text{a}} - T_{\text{r}}}{T_{\text{a}} - T_{\text{wetbulb}}}. \tag{26}$$

The air temperature leaving the cooling device can be calculated from (26)

$$T_{\text{r}} = T_{\text{a}} - \eta \, \text{Saturation}(T_{\text{a}} - T_{\text{wb}}). \tag{27}$$

The wet bulb temperature of the air entering and exiting the device is assumed to be constant, where the minimum outlet temperature equals the inlet wet bulb temperature.

Since the performance in the evaporative cooler is an adiabatic process,

$$Q_{\text{Latent}} = Q_{\text{Sensible}}. \tag{28}$$

The humidity ratio of the air outlet from the device [22]

$$\text{RH}_{\text{out}} = \frac{X_{\text{inlet}} p_{\text{outside}}}{(0.622 + X_{\text{inlet}}) p_{\text{saturated.vapour.pressure}}}. \tag{29}$$
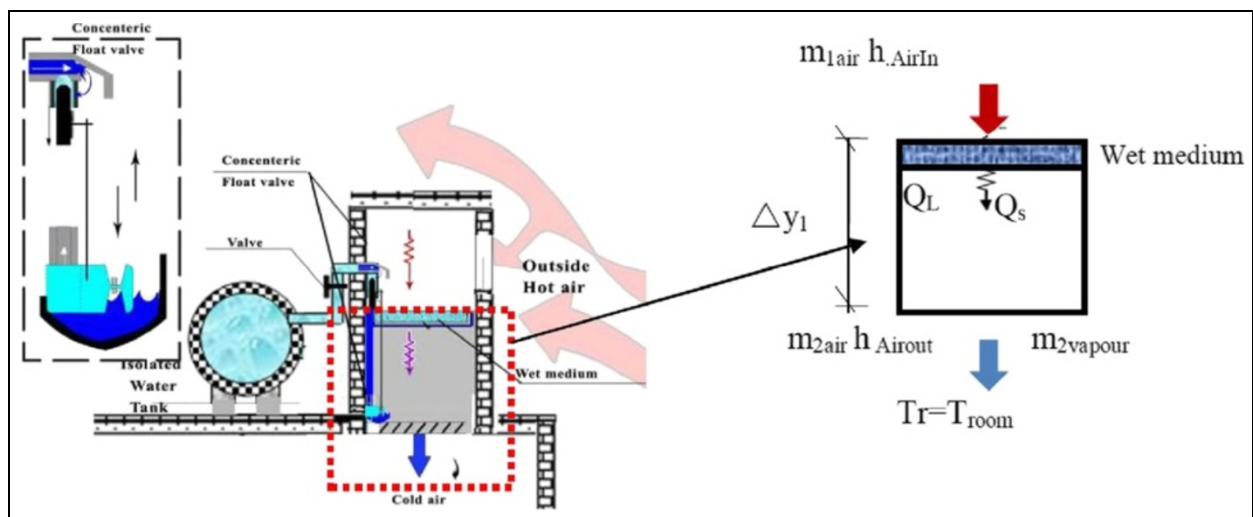
Figure 6 Evaporative cooler schematic.

The saturate vapor pressure (atm). Bouchahm, 2011 [23] is

$$P_{\text{saturated.vapour.pressure}} = 10^{\left(17.443 - \frac{2795}{T-273} - 3.868\log_{10}(T-273)\right)}. \tag{30}$$
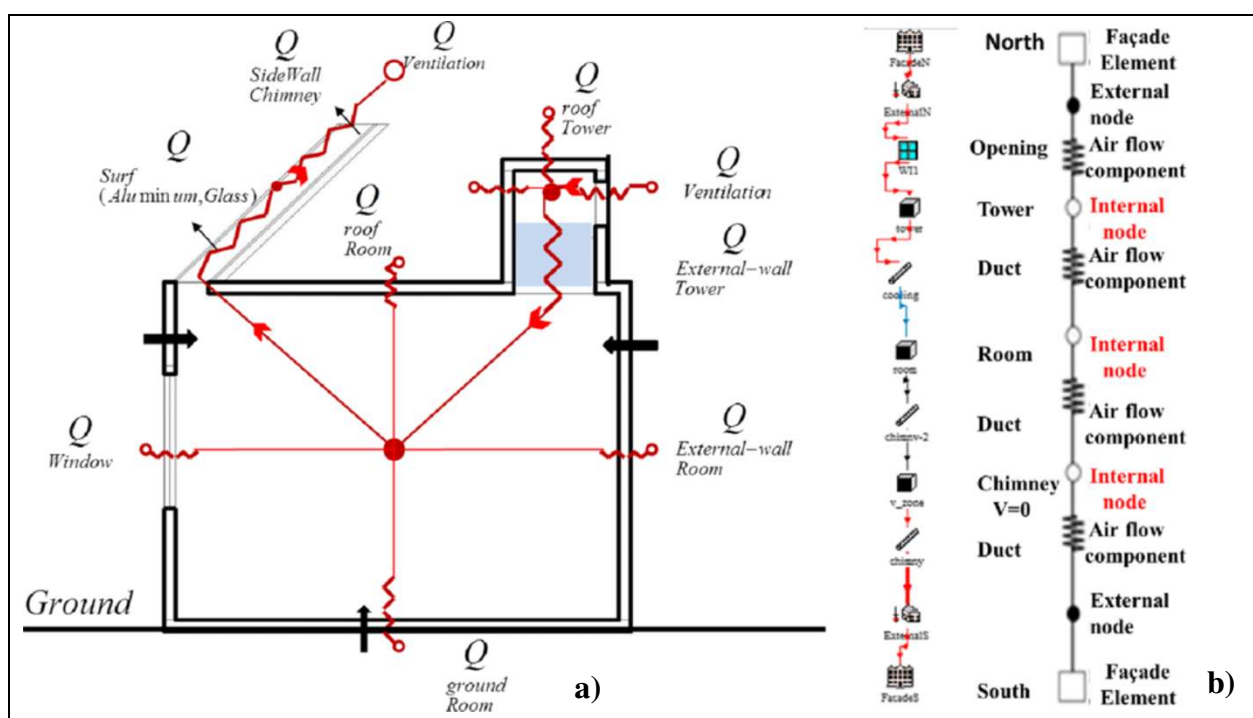
The corresponding energy removed from the air stream to evaporate water under these circumstances is given by Equation 31:

$$Q_{\text{air}} = \dot{m}_{\text{cooler}}(h_{.\text{AirIn}} - h_{.\text{AirOut}}), \tag{31}$$

where $h_{\text{enthalpy}} = C_p\, T + x(r + C_{\text{P.vapor}}T)$, expressed in kJ/kg (dry) [24]:

$$1.005\, T + x(2501.1 + 1.846T), \tag{32}$$

where $C_{\text{p.air}}$ = the specific heat of air (kj/kg.K), x = humidity ratio (kg/kg (dry)), r = evaporative latent heat,

Figure 7 The building thermal model and the air flow network of the integrated model. (a) Heat flows in the ECWT, room, and solar chimney with combination of ventilation rate in TRNSYS model; **(b)** air flow network in COMIS with a description of the internal node, external node, and airflow components.

$C_{p.vapor}$ = the specific heat of vapor, and T = temperature (°C).

Air flow network and building thermal model

The ventilation model is integrated to the conventional building thermal model in order to have strong influence on the system performance. The internal heat load, solar load, and the heat transfer through walls, windows, roof, ground, and opening are considered and shown in Figure 7a, where Q is the heat transfer. The temperature of the ground at depth of 0.5 m is used in calculation for heat transfer from the ground and derived from Egyptian Typical Meteorological Year (ETMY) which is developed for standards development and energy simulation by US National Climatic Data Center (CA, USA) for a period of 12 years from 1991 until 2003 [25]. Also, the internal load is used in the calculation, where there are four occupants inside the single zone doing light work and seating with light load. Each occupant is assumed to have a sensible gain of 75 W and latent gain of 75 W according to ISO 7730 [26]. The occupants are assumed to be staying in the room for the whole day to apply the maximum load inside. The room light has 13 W/m². It has an energy saving lamp with convective part (visible) of 20%, and one device of 140 W is used.

The zones (ECWT, room, and SC) are linked to their ambiance via external nodes and to the north and south facades. Wind pressure data, wind speed, wind velocity profile for the building and for meteo, and orientation are defined. The COMIS program uses wind speed at meteo site in reference height to calculate the speed above the boundary layer using the wind velocity profile of the meteo station. The boundary layer is 60 m high for smooth terrain. Wind pressure coefficients (Cp values) belonging to a certain façade element can be given for several wind directions derived from 'CP-Generator' [13,27] according to the room with the integrated system with maximum height of 4.5 m and around the building. Figure 7b shows the air

flow network in COMIS. Table 2 shows the building construction material description based on the ideal thermophysical properties [28] which was used in TRNBuild. The building thermal model is calculated based on the mathematical Equations 33 to 38.

The heat transfer through external walls (and roof) for room zone and tower zone is predicted as

$$\sum_{\text{external}} Q = \sum^{ne} \frac{A_z(T_{\text{star}} - T_z)}{R},\tag{33}$$

where ne is the number of external walls attached to the zone, T_{star} is star temperature which can be used to calculate a net radiative and convection heat flux from the wall surface, R is the overall thermal resistance value including convective heat transfer resistance on both sides of the surface, and z is the zone [26].

The total gain to chimney zone from all surfaces is the sum of the combined heat transfers:

$$\sum_{\substack{\text{Surface}\\\text{Chimney}}} Q = \sum A_s \dot{Q}_{\text{Comb}\_z} + S_{\substack{\text{outside}\\\text{surface}}},\tag{34}$$

where A_s is the inside area of the surface, and $S_{\text{outside surface}}$ is the solar radiation for outside surface.

The heat transfer across the windows is

$$\sum_{\text{Window}} Q = \sum A_z U_W(T_a - T_{\text{star}}) + I(\text{SHGF}),\tag{35}$$

where U_w is the overall heat transfer coefficient across the window (the rate of heat transfer through the window from outside to inside when it is hot per unit area and per unit temperature difference (W/m²C), SHGF is

Table 2 Building material description used in calculation

Building part	Material part	Conductivity (kJ/h m⁻¹ K⁻¹)	U-value (W/m²K⁻¹)	Thickness (m)
Glass windows	Single glass	-	5.68	0.004
External walls	Common plaster (coating)	1.26	2.60	0.02
	Brick	3.60		0.10
	Common plaster (coating)	1.26		0.02
Roof	Insulation	0.2	0.443	0.05
	Concrete slab	4.2		0.12
	Cement plaster (coating)	4.50		0.01
Ground	Floor	-	0.797	0.10
	Insulation	0.2		0.02
	Concrete	4.2		0.10

Table 3 The results of the numerical simulation after using the new proposed system

Input			Outputs after cooling			
Ambient temperature (°C)	Solar radiation (W/m^2)	Ambient RH (%)	Indoor temperature (°C)	Indoor RH (%)	Indoor ACH	Indoor (m^3/h)
34	1,016	17	34	52.23	6.93	346.84
		25	25.64	58.65	6.95	347.75
38	1,016	17	27.45	51.39	6.80	340.36
		25	28.91	57.97	6.82	341.29
42	1,016	17	30.52	50.63	6.68	*334.05*
		25	32.13	57.53	6.70	335.09
34	425	17	24.33	52.36	5.45	272.95
		25	25.59	58.82	5.48	274.33
38	425	17	27.41	51.51	5.34	267.05
		25	28.83	58.22	5.37	268.53
42	425	17	30.48	50.74	5.22	261.39
		25	32.08	57.69	5.25	262.97
34	0	17	24.20	52.78	4.28	214.15
		25	25.45	59.31	4.31	215.87
38	0	17	27.28	51.91	4.17	208.75
		25	28.70	58.69	4.21	210.54
42	0	17	30.36	51.11	4.07	*203.57*
		25	31.95	58.14	4.10	205.45

the window's solar heat gain factor and its calculation is presented in details [29].

For external surfaces, the long wave radiation is considered using a fictive sky temperature T_{sky} [23]:

$$\dot{Q}_{Comb} = \dot{Q}_{convective} + \dot{Q}_{radiative} \tag{36a}$$

$$\dot{Q}_{convective} = h_{ConvOutside}\left(T_{AdjSurface} - T_{outsideSurface}\right) \tag{36b}$$

$$\dot{Q}_{Radiative} = \beta\varepsilon_{outside}\left(T_{OutsideSurface}^4 - T_{fsky}^4\right) \tag{36c}$$

$$T_{fsky} = \left(1 - f_{sky}\right)T_{AdjSurface} + f_{sky}T_{sky}, \tag{36d}$$

where f_{sky} is the view factor to the sky, and ε is the long wave emissivity of outside surface = 0.9 for walls.
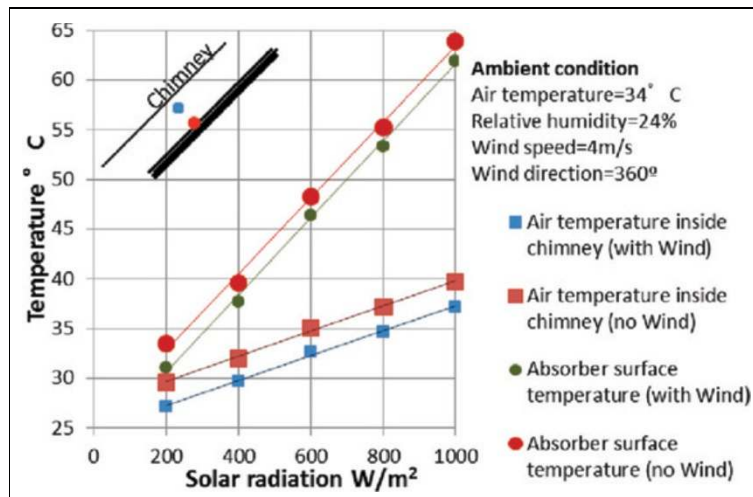
Figure 8 The variation of the absorber and chimney temperature with solar radiation.
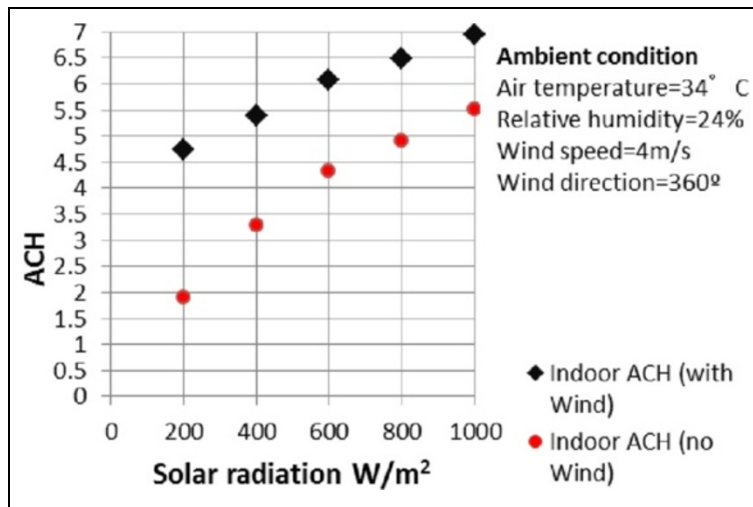
Figure 9 The variation of the ACH with solar radiation with different combinations.

The sky model used to calculate the radiation on inclined surface for the solar chimney by Perez model, which is considered the best available model.

Heat transfer due to ventilation in a zone is

$$\sum_{\text{Ventilation}} Q = \sum_{\text{Infiltration}} \dot{m}\, c_{\text{p}} T_{\text{outside}} - \sum_{\text{FromZone}} \dot{m}\, c_{\text{p}} T_{\text{zone}}$$
$$+ \sum_{\text{ToZone}} \dot{m}\, c_{\text{p}} T_{\text{zoneAdjacment}}. \qquad (37)$$

Then, the total heat load to maintain the zone at specified temperature can be calculated as

$$Q_{\text{load}} = \sum_{\text{Ventilation}} Q_{\text{Out}} + \sum_{\substack{\text{External \_wall} \\ \text{Tower}}} Q + \sum_{\substack{\text{External \_wall} \\ \text{Room}}} Q + \sum_{\substack{\text{ground} \\ \text{Room}}} Q + \sum_{\substack{\text{Window} \\ \text{Room}}} Q$$
$$+ \sum_{\substack{\text{Roof} \\ \text{Tower}}} Q + \sum_{\substack{\text{Roof} \\ \text{Room}}} Q + \sum_{\substack{\text{Side \_wall} \\ \text{Chimney}}} Q + \sum_{\substack{\text{Surface} \\ \text{Chimney}}} Q + \sum_{\substack{\text{People} \\ \text{Light}}} Q. \qquad (38)$$

Then, the zone temperature is iteratively predicted in the calculation.

Results and discussions
Steady state condition
The performance of the integrated system has been numerically studied, and the effect of the pressure coefficient on ACH and system performance was studied during the steady state condition. The investigation was at constant wind speed at 4 m/s (at Meteo), wind direction of 360°, and pressure coefficient (C_{p}) 0.081 for the north façade, and −0.040 for the south façade as shown in Table 3. Also, the effect of the different combinations of the three values of solar radiation, temperature, and relative humidity on indoor environment was studied. The main results show the increase of ACH with an increase of solar radiation. The system is capable of generating 130.5 m³/h ventilation rates for a collector area of
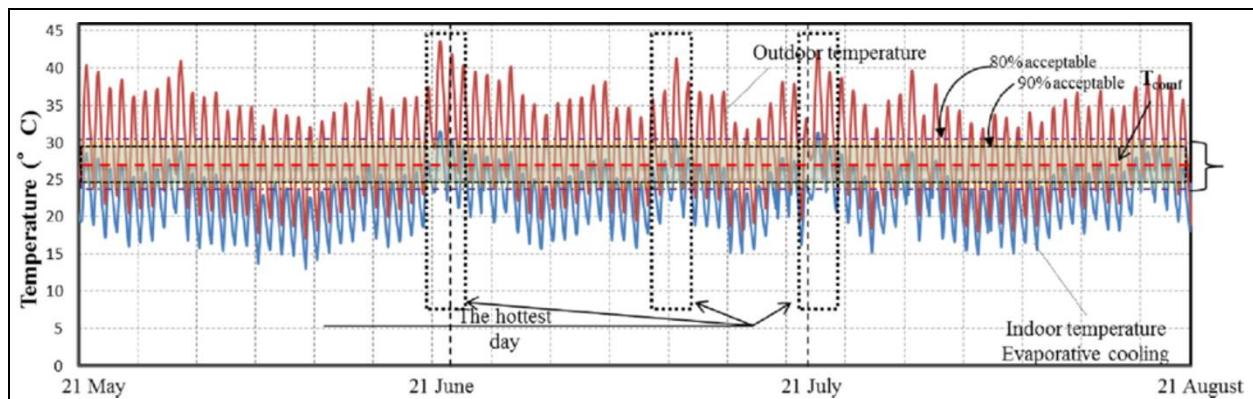
Figure 10 The performance of the system during 3 months of summer using standard weather data.
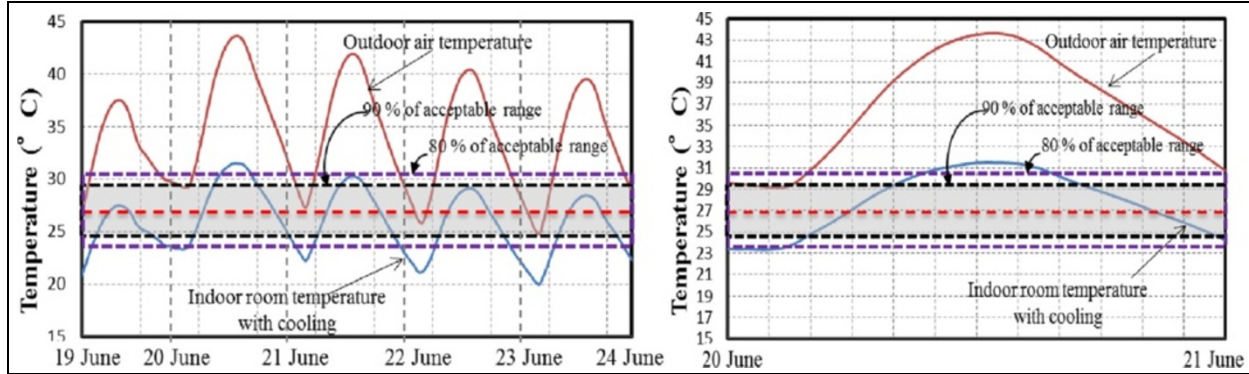
Figure 11 The performance of the system on 19 to 23 June with focus on the hottest day of June.

2.4 m² under the effect of solar radiation only. The values of these induced air flows depend on the geometry of the air collector and the performance parameters of the solar collector.

It can be seen from Figure 8 that the chimney cavity temperature is always higher than the ambient temperature with a very high temperature for an aluminum surface (black absorber) that reaches to 64°C at the radiation level of 1,000 W/m², then the surface temperature decreases to 40°C at the radiation level of 400 W/m². It has been observed that the black absorber possesses a high temperature value compared to chimney air temperature; this increase in temperature is due to more captured radiation and storage of high thermal energy.

This absorbed energy accelerates the flow of air through the chimney during the day and the night. As a result, this exit air flow will help more fresh air to enter to the space from the opening of the wind tower. Figure 9 shows the increase of ACH with an increase of solar radiation and pressure coefficient with maximum 7 ACH inside the room. Also, the minimum indoor air change rate is 2 ACH with 200 W/m² of solar radiation and no wind effect. Thus, the

indoor ventilation rate reached zero with no radiation and no wind pressure coefficient. The temperature of the air is higher at the inlet and decreases at the outlet in the evaporative cooler (EC). This is because, initially, the dry air has a large potential to absorb the moisture and gets saturated as the airflow reaches the outlet of the ECWT.

According to ASHRAE 62–2001 standard (Atlanta, GA, USA), ventilation rates dependent on the floor area, whereas the minimum ACH is 0.35, but no less than 15 CFM/person. Also, the Egyptian Energy Code for residential buildings implies a minimum natural ventilation rate of 3 L/s-person (10.8 m³/h) for the living room [30]. Therefore, the developed system is able to provide the desired ventilation and thermal needs of the occupants with/without solar radiation as shown in Table 3.

Real weather data

The real weather data of the New Assiut City which is taken from ideal weather data of 20 years [26] was applied to study the system performance during the summer season and the hottest days. In terms of thermal comfort, ASHRAE recognizes that the conditions
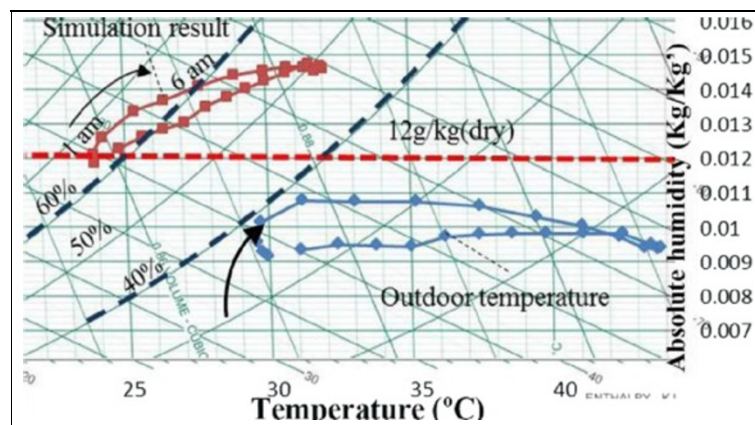
Figure 12 The humidity environment of the hottest day (20 June) using psychrometric chart of ASHRAE. Blue: outdoor temperature. Red: indoor temperature.

required for thermal comfort in spaces that are naturally conditioned are not necessarily the same as those required for other indoor spaces [31]. So, a new ACS for a naturally ventilated building has been proposed to be integrated within ASHRAE Standard 55, 2004. In the ACS, the mean monthly outdoor air temperature determines the acceptable indoor operative temperature. This relationship is expressed by the following formula: $To_{(comf)} = 0.31Ta_{(out)} + 17.8$, where $To_{(comf)}$ is the optimum comfort operative temperature in °C and $Ta_{(out)}$ is the mean monthly outdoor air temperature in °C.

Therefore, most of the outlet temperature after passing through the wetted medium in ECWT is below the acceptable limits of Adaptive Comfort Standard (ACS) as shown in Figure 10. Only 5% of the results data exceed the upper limit of 80% acceptable range of ACS especially for the hottest days. An investigation of the system performance during the 5 days of June was done to focus on the hottest day in the summer season from the weather data file so that a clear pattern is seen in relation to adaptive comfort standard (ACS) as shown in Figure 11.

Arundel et al. proposed that the low or high humidity strongly affects the health and the optimum humidity level which is between 40% and 60% [32]. Therefore, the humidity environment in the living room and the bedroom is very important. Figure 12 shows most of the simulation results for the hottest day, which are located between 40% and 60% except before sunrise between 1 am and 6 am. Thus, this shows the effectiveness of a natural ventilation strategy with a direct evaporative cooling for an indoor environment. Finally, this proposed compact system could be an economical and practical passive alternative to the conventional air conditioning systems in hot and dry climates with no pollution released or energy consumed.

Conclusions

Important conclusions are drawn from the numerical operations of a new integrated system. Pressure coefficient and solar radiation in relation to the area of the collector are the main parameters that affect the air flow rate inside a single zone. When the temperature of the black absorber increases with/without pressure coefficient effects, the ACH increases with a strong relation to solar radiation as shown in Figure 8. The temperature of the indoor environment decreases by 10°C to 11.5°C compared to the outdoor temperature. This helps to achieve comfort during the hottest days of the summer season, with 95% of indoor temperature below the upper range of 80% acceptable comfort range during the period from 21 May to 21 August. The proposed system can be applied during daytime and nighttime, but it can be controlled and its use can be limited during nighttime especially when the inlet outdoor temperature is ≤30.5°C.

Overall, the results show how passive systems interact with the building and weather conditions. This paper studied the effect of a new integrated system for indoor comfort for the New Assiut City as a case study. The authors' future work will be a parametric investigation and optimization of the important operating parameter to achieve compact and high performance design so that guidelines for an efficient optimization process will be done to optimize cases with different locations, dimensions, and number of occupants.

Nomenclature
The list of nomenclature is presented in Table 4.

Table 4 Nomenclature

Symbol	Description	Unit
v	Air velocity	m/s
Q	Volume flow	m³/s
T_{Wb}	Wet bulb depression	°C
Q_{air}	Energy transferred to or removed from the air stream	kJ/h
$\dot{m}_{coolere}$	Mass flow rate of the air through the cooling device	kJ/h
$\dot{m}$	Mass flow rate of the air	kg/s
h_{AirIn}	Enthalpy of air entering the device	J/kg
h_{AirOut}	Enthalpy of air exiting the device	J/kg
T	Temperature	K
t	Temperature	°C
ΔT	Wet bulb depression DRT-WBT	°C
ΔT_{LM}	Log mean temperature difference	°C
C_p	Specific heat at constant pressure	kJ/kg °C
$C_{psouth,north}$	Wind pressure coefficient	-
C_f	Specific heat of air	J/kg K
Δw	Humidity ratio	-
S_g	Solar radiation absorbed by glass	W/m²
S_w	Solar radiation absorbed by absorber	W/m²
h_{rwg}	Radiative heat transfer between absorber and the glass	W/m² K
h_{rs}	Radiative heat transfer between the glass and the sky	W/m² K
h_g	Convective heat transfer between glass and sky	W/m² K
h_{wind}	Convective heat transfer outside glass due to wind	W/m² K
h_w	Convective heat transfer between the absorber and the air	W/m² K
T_a	Temperature of ambient air	K
T_{sky}	Sky temperature	K
T_g	Mean glass temperature	K
$T_{f,air}$	Temperature of air in the chimney	K

Table 4 Nomenclature *(Continued)*

T_w	Temperature of the absorber	K
T_{room}	Room temperature	K
q''	Heat transfer to the air stream in the chimney	w/m^2
U	Overall convective heat transfer coefficient	$W/m^2\,K$
V	Wind speed at reference height	m/s
I	Incident solar radiation	W/m^2
P	Atmospheric pressure	pa
$\dot{Q}$	Total heat transfer	$W/m^2\,K$
G	Gravitational constant = 9.8	m/s^2
k	Thermal conductivity	$W/m\cdot K$
Dimensionless terms		
Nu	Nusselt number ($h_f d/k_{air}$)	
Ra	Rayleigh number	
Greek symbols		
ε	Emissivity of glass = 0.9, emissivity of absorber = 0.95	
γ	Temperature weighting factor = 0.75	
$\eta_{Saturation}$	Saturation efficiency of the evaporative cooling	
ρ	Air density (kg/m^3)	
τ	Transmissivity	
Subscripts		
a	Ambient	
g	Glass	
w	Absorber	
Cav	Chimney cavity	
r	Room	

Competing interests

The authors declare that they have no competing interests.

Authors' contribution

All authors read and approved the final manuscript.

Authors' information

ASHA graduated in 2004 and received his M.S. in 2009 from the Department of Architecture, Assiut University, Egypt. He spent 2 years (2011 to 2013) as a researcher in Department of Architecture and Building Science, Tohoku University, Japan, for his of Ph.D. degree. YH is an Emeritus Professor, Department of Architecture and Building Science, Tohoku University. TG is an Associate Professor in the Department of Architecture and Building Science, Tohoku University. NE is a researcher, Department of Architecture and Building Science, Tohoku University. MMR is a Professor in the Department of Architecture, Assiut University. MAE is a professor in Department of Architecture, Assiut University.

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