AMERICAN UNIVERSITY OF BEIRUT

PASSIVE COOLING SYSTEM AIDED WITH PERSONAL EVAPORATIVE COOLING AND DESICCANT-TROMBE WALL DEHUMIDIFICATION FOR HOT HUMID CLIMATE.

by

MOHAMMAD JAMAL EL LOUBANI

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submitted in partial fulfillment of the requirements
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to the Department of Mechanical Engineering
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Approved by:

Dr. Kamel Abou Ghali, Professor Mechanical Engineering	Advisor
Dr. Nesreen Ghaddar, Professor Mechanical Engineering	Co-Advisor
Dr. Fadl Moukalled, Professor Mechanical Engineering	Member of Committee

Date of thesis defense: December 6, 2019

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AN ABSTRACT OF THE THESIS OF

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Title: Passive Cooling System Aided with Personal Evaporative Cooling and Desiccant-Trombe Wall Dehumidification for Hot Humid Climate

This study investigates the performance of a hybrid cooling system for an office space. The hybrid system combines a phase change material (PCM) storage layer with a melting temperature of 25 ºC to cool the supply air and a personalized evaporative cooler (PEC) system to provide local cooling for the occupants. In addition, humidity of the supply air is controlled using a solid desiccant wheel regenerated via an auxiliary heater assisted with a Trombe wall. Moreover, an evaporative cooler and a sensible wheel are used to assist in cooling the supply air. The combined system can be used primarily for hot humid climates in office spaces. The system should meet the space load, indoor air quality, and minimum fresh air requirements (7 *l*/s per person) set by ASHRAE standards and can provide thermal comfort during daytime with minimum electric energy consumption by using only small fans and an auxiliary heater. Mathematical models for each component have been implemented to simulate their performance and then evaluate the performance of the overall system. A bioheat model that can predict human thermal response such as skin and core temperatures has been used to predict the overall thermal comfort of the occupants.

A case study for an office space located in Beirut is developed to assess the performance of the proposed hybrid cooling system during the summer months and to check the resultant overall thermal comfort of the occupants. The Trombe wall proved to be effective for regeneration purposes as it achieved a 55 % thermal energy reduction compared to relying only on the auxiliary heater for regeneration. Overall, the proposed system achieved savings up to 87% of the total operation cost compared with the consumption of a conventional AC unit over the entire cooling season.

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NOMENCLATURE

A : area, m^2

C_p : specific heat, J/kg·K

f : melting fraction

h : convective heat transfer coefficient, $W/m^2 \cdot K$

H : enthalpy, kJ/kg

h_{sf} : latent heat of fusion of the PCM, J/kg

I : solar radiation, W/m^2

k : thermal conductivity, W/m·K

$\dot{m}$: mass flow rate, kg/s

PF : performance factor

Q : power, W

T : temperature, °C

t : time, s

U : overall heat transfer coefficient, $W/m^2 \cdot K$

V : volume, m^3

w : humidity ratio, kg/kg

Greek symbols

α : absorptivity

ε : emissivity

ρ : density, kg/m^3

τ : transmissivity

ϕ : relative humidity, %

Subscripts

amb : ambient

ch : channel

cond : conduction

conv : convection

EC : evaporative cooler

eq : equipment

g :glazing

occ : occupants

PEC : personalized evaporative cooler

pcm : phase change material

r : room

rad : radiation

reg : regeneration

sup : supply

W : wall

wb : wet bulb temperature

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CHAPTER I

INTRODUCTION

Conventional heating, ventilation and air conditioning (HVAC) systems are widely used for cooling or heating purposes, thermal comfort provision, and indoor air quality improvement in buildings [1]. However, these systems consume a large amount of energy amounting to nearly 40% of the total building energy consumption [2]. In addition, they negatively impact the environment accounting for more than 30% of the carbon dioxide emissions [2]. They also disrupt the energy supply and demand balance in warm and humid climates [3]. Therefore, sustainable and passive alternative techniques are primarily sought to reduce the use of conventional HVAC systems [4]. One of these passive cooling techniques is the free cooling of buildings [5]. Free cooling ventilation is defined as using the outdoor air as a heat sink for building cooling purposes [6,7]. Direct free cooling can be implemented in moderate climates when the outdoor air temperature is within the thermal comfort range of temperatures. However, in hot climates, the outdoor air temperature is higher than the thermal comfort level. To benefit from free cooling in these climates, the natural cold energy available at night due to the outdoor air temperature difference between the day and the night should be utilized [8]. Thus, a storage system would be needed to collect this energy and use it during the warm period of the following day. A sustainable passive storage system that can be utilized for this purpose is known as cool thermal energy storage [9,10].

Thermal energy storage is a passive technique that can preserve energy when it is available to be extracted when it is needed [1]. One of the thriving techniques to store thermal energy is the implementation of phase change materials (PCMs) [9]. PCMs are characterized by their ability to store and release large amounts of energy nearly isothermally [11]. PCM can be installed into the building in passive systems through integrating them into the building envelope (walls, roofs, and floors) [12]. It was found that PCM integrated in buildings is effective in climates where the temperature difference between the day and the night is large [8]. Osterman et al. [13] conducted an experiment of integrating PCM in a building in Slovenia, and their findings were that the maximum cooling potential was in August and July when there were more fluctuations in temperature between day and night. However, there are many climates where the temperature difference between day and night is small such as the Mediterranean coastal areas including the Lebanese coast. Therefore, to be able to benefit from thermal energy storage systems, means of cooling the outdoor air are needed such as evaporative cooling coupled with selecting PCMs at a moderate melting temperature. However, selecting a PCM with a moderate melting temperature may not be low enough to provide room air temperature within thermal comfort levels. So, another passive strategy should be employed in addition to the PCM to assist in cooling the occupied space.

A promising passive system that can assist the PCM in providing thermal comfort is the personalized evaporative cooler (PEC). The concept of personalized ventilation relies on transporting and delivering the conditioned air directly to the occupant face and chest [14,15]. Zhang [16] showed that upper body parts have a dominant impact on the overall sensation while cooling. Thus, the overall thermal comfort of the occupant was improved without the need for having homogenous cool air in the occupant's whole-body microclimate. This permitted the use of elevated supply air temperatures to the space resulting in higher background or macroclimate room temperature [17-19]. Consequently, the PEC would be attractive to use in passive applications where the background room air temperature is at the high end. The PEC can cool the area around the person while only consuming a small amount of water and electrical energy for operating the fan. Cooling is done by water evaporation which can reduce the temperature of the air when the air is relatively dry [20]. However, a strategy is needed to dehumidify the room air so that the PEC can be employed in humid climates of coastal regions.

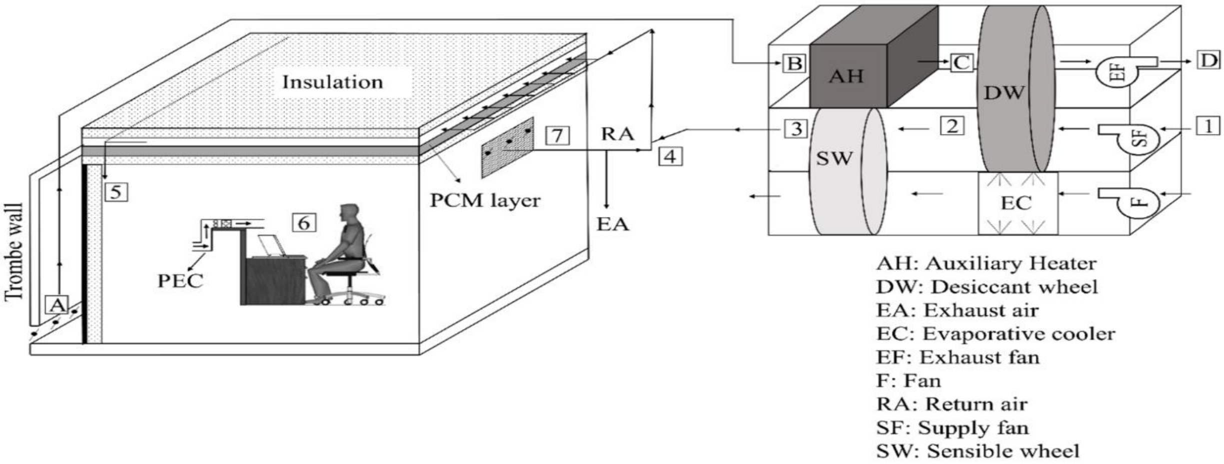
For this purpose, desiccant dehumidification is an attractive option in these applications when outside air is humid [21]. Solid desiccant wheels are widely employed as a dehumidification system and can be regenerated using a low-grade regeneration heat source [21]. The heat source could be an auxiliary heater or a free source of heat. To reduce energy consumption of the auxiliary heater, a Trombe wall can be integrated with the building south envelope to heat an air stream between the glazing and the wall and use this air to regenerate the desiccant. The Trombe wall was reported in literature to heat the ambient air substantially. In a study conducted by Bevilacqua et al. [22], the Trombe wall heated the outdoor air from 11 °C to 23 °C during the winter season in Pisa. Also, Hong et al. [23] reported an increase of the ambient air temperature from 5 °C to 22 °C when the Trombe wall was used in the winter season in China. So, the Trombe wall can be utilized to reduce the energy consumption of heating, as it was mainly used for heating purposes during winter. The use of the Trombe wall for regenerating desiccants has not been reported in previous studies.

In this study, a hybrid passive cooling system is proposed combining a PCM storage layer, a PEC, and a solid desiccant wheel regenerated via an auxiliary heater aided with a Trombe wall. This system can be implemented in hot and humid regions such as the Mediterranean or the Gulf coastal regions. The proposed system is applicable for office spaces which are unoccupied during the night to allow for thermal energy storage by the PCM. The proposed system supplies a minimum amount of fresh air according to ASHRAE standards (7 *l*/s per person) [6] and can provide thermal comfort with minimal electric energy divided between the energy of operating small fans and an auxiliary heater. Moreover, the proposed system consumes a minimal amount of water for operating the evaporative cooler and the PEC. Mathematical models of each component are developed to predict the performance of the overall integrated system. The thermal comfort and the overall energy consumption of the system are assessed and are compared with the conventional HVAC system.

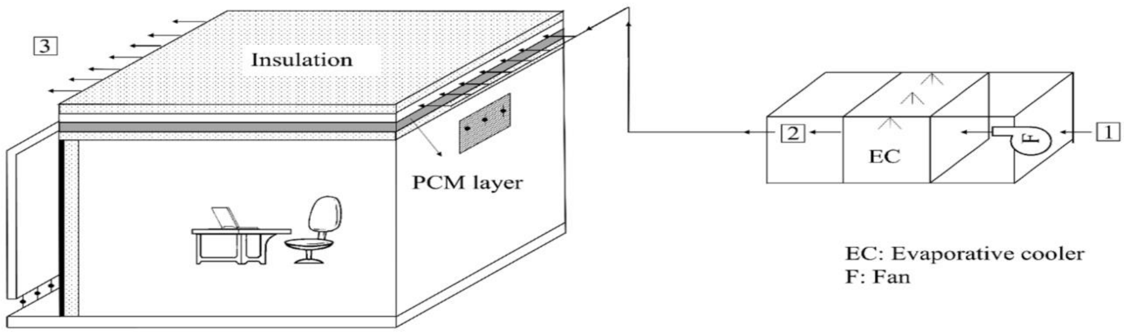
CHAPTER II

SYSTEM DESCRIPTION

The proposed system has 2 modes of operation: daytime and nighttime operation as shown in **Fig. 1** (a) and (b). The main components of the system are: a PCM storage layer system, PEC, a desiccant wheel, a Trombe wall, an auxiliary heater, an evaporative cooler, and a sensible wheel. The proposed operation of the integrated system is described in this section.



(a)



(b)

Fig. 1: Schematic of the proposed system during (a) daytime and (b) nighttime operation

A. Daytime Operation

Outdoor air at (1) is dehumidified after passing through the desiccant wheel to reach (2). The air gains heat after the dehumidification process, so with the aid of evaporative cooling, the ambient air cools the dehumidified air as it passes through the sensible wheel to reach (3). The needed fresh air mixes with the room return air that will recirculate in the space at (4). After cooling the mixed air stream through exchanging heat with the PCM storage layer, the air is supplied to the space at (5). The PEC then draws the room air and supplies the face and the trunk of the occupant with the evaporatively cooled air at (6). Finally, part of the room air is exhausted to the ambient while the other part at (7) is mixed with the fresh outdoor air and the process repeats throughout the day. As for the regeneration of the desiccant wheel, first the Trombe wall heats the outdoor air at (A) to reach (B). Then, the air passes through the auxiliary heater at (C) which will heat the air to the desired regeneration temperature. As the air leaves the auxiliary heater, the air passes through the desiccant wheel to regenerate it, and the air exiting the wheel at (D) is then exhausted to the ambient.

B. Nighttime Operation

The operation of the system during nighttime is shown in **Fig. 1(b)**. A control strategy is implemented to supply fresh outdoor air before 3 hours of the beginning of the occupancy schedule to refresh the room air with the cooler and less humid outdoor air available during the early hours of the morning. As for the remaining unoccupied hours, a small flow of outdoor air enters the space due to infiltration. The nighttime operation is also used to regenerate the PCM from the cool energy available at night. A control strategy is implemented to optimize the regeneration process when the ambient air is only admitted to the PCM air channel if its dry bulb temperature is less by 1.5 °C or more than that of the PCM melting temperature. Otherwise, evaporative cooling is used to aid the regeneration process of the PCM during the unoccupied hours. Evaporative cooling at best can reduce the ambient air temperature to its wet bulb value. Thus if the temperature of the air leaving the evaporative cooler is below the PCM melting temperature, the outdoor air is drawn to the evaporative cooler which cools it from (1) to (2). The air then cools the PCM layer and leaves with a higher temperature at (3). When complete regeneration is achieved and the PCM temperature at the end of the channel is 1 °C less than the PCM melting temperature, both the evaporative cooler and the fan used to circulate the air will not be used, and no ambient air is introduced to the PCM air channel.

CHAPTER III

MATHEMATICAL MODELING

To simulate the performance of the proposed hybrid passive cooling system, each subsystem was modeled using a mathematical model adopted from published validated models in literature; i) space model [24,25], ii) PCM system model [26], iii) PEC [27] and thermal comfort model [28], and iv) desiccant model with its regeneration system that uses a Trombe wall and an auxiliary heater. The models were integrated to find the conditions of the supply air, the room air, and the air reaching the occupant. In addition, the thermal comfort of the occupants was assessed using a segmental bioheat model and a comfort model suitable for non-uniform environments around the occupant. In what follows, the mathematical models of the subsystem are described.

A. Space Model

Assuming that the room air is homogeneous and that well mixed conditions apply, the lumped energy and mass balance equations for the office space are presented below respectively:

$$\rho_{air} V_r C_p \frac{dT_r}{dt} = \dot{m}_{sup} C_p \left(T_{sup} - T_r\right) + Q_{occ} + Q_{eq} + Q_{conv} \tag{1}$$

$$\rho_{air} V_r \frac{dw_r}{dt} = \frac{Q_{latent}}{h_{fg}} - \dot{m}_{sup} \left(w_r - w_{sup}\right) - \dot{m}_{PEC} (w_r - w_{PEC}) \tag{2}$$

Where the left side of Eq. (1) represents the transient heat stored in the room air; the first term of the right side is the net convective heat flow; Q_{occ} is the sensible load generated by the occupants; Q_{eq} is the sensible load generated by the lights and other electrical equipment inside the space; Q_{conv} is the convective heat transfer between the room air and the envelope of the space.

The left side of Eq. (2) represents the transient moisture stored in the room air; the first term of the right side is the moisture generation rate in the room; the second term of the right side corresponds to the net convective moisture flow; the last term of the right side represents the moisture flow from the air leaving the PEC nozzle to the room air.

The walls' surface and internal temperatures were found using the space model presented by Yassine et al. [24] and Hourani et al. [25] where 1D transient conduction is assumed, and each wall is divided into parallel layers. The temperature across these layers is found while taking into account the ambient and indoor conditions of the boundary layers of each wall.

B. PCM Storage Model

The schematic of the PCM storage layer system is shown in **Fig. 2**. This system is decomposed of a single PCM layer and a parallel single air channel above it, both sandwiched between insulation layers.

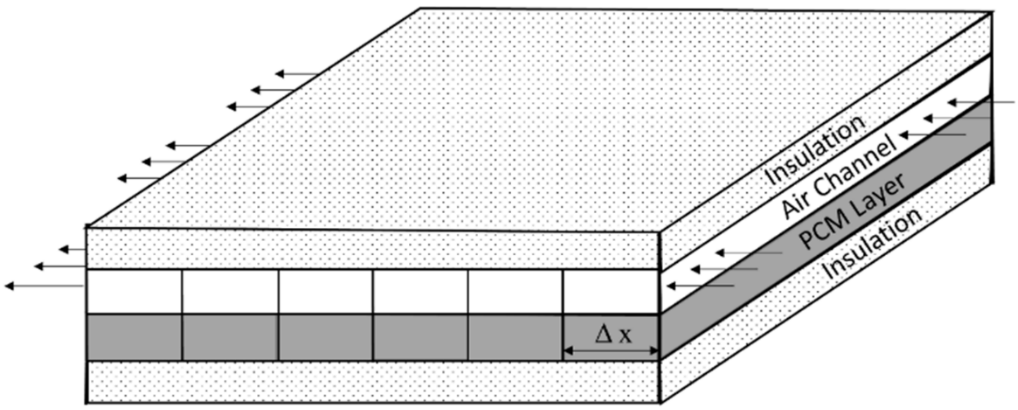


Fig. 2: Schematic of the PCM storage system

The model of Lin et al. [26] was adopted in this work which was experimentally validated. The model used the enthalpy method to simulate the PCM performance while assuming 1-D transient heat transfer. Moreover, the PCM layer was made up of PCM bricks and the air channel was divided into elements of dimensions equal to that of the PCM brick, so that each element has a uniform temperature (see **Fig. 2**). In addition, the conduction between two adjacent PCM bricks was assumed negligible due to the high contact resistance between the bricks. The ceiling of the occupied space was also assumed to be well insulated, so there was no heat transfer between the PCM and the ceiling.

The energy balance equation at any time *t* for each element *i* of the air channel is as follows:

$$\rho_{air} V_i C_p \frac{dT_{air,i}}{dt} = h_{air-pcm} A_i \left(T_{PCM,i} - T_{air,i}\right) + \dot{m}_{sup} C_p \left(T_{air,i-1} - T_{air,i}\right) \tag{3}$$

Where the left side represents the transient heat storage in the air channel; the first term of the right side constitutes the convective heat transfer between the PCM and the air layer; the last term of the right side corresponds to the net convective heat flow; $T_{air,i}$ and $T_{PCM,i}$ are the temperatures of the considered air and PCM elements respectively; $T_{air,i-1}$ is the temperature of the element just before the considered element.

As for the PCM layer, the heat exchange is defined for two modes: 1) sensible heat exchange and 2) phase change heat exchange. In the sensible heat exchange, the PCM layer exchanges sensible heat with the air without any PCM phase change occurring and during which the PCM temperature is different than its melting temperature. The energy balance equation that was used to solve for the PCM temperature is the following:

$$\rho_{pcm} V_i C_{p-pcm} \frac{dT_{pcm,i}}{dt} = h_{air-pcm} A_i \left(T_{air,i} - T_{PCM,i}\right) \tag{4}$$

Where the left side of the equation represents the transient heat storage in the PCM layer, and the right side is the convective heat transfer between the PCM and air layer.

In the phase change heat exchange, where the PCM temperature is equal to its melting temperature, the PCM undergoes an isothermal heat exchange until it completely solidifies or completely liquefies. In this case, the melting fraction of the PCM element can be found using the following energy balance equation:

$$\rho_{pcm} V_i h_{sf} \frac{df_i}{dt} = h_{air-pcm} A_i \left(T_{air,i} - T_{PCM,melting}\right) \tag{5}$$

The left side of the equation represents the transient heat storage in the PCM layer; the right side of the equation corresponds to the convective heat transfer between the PCM and air layer; h_{sf} is the latent heat of fusion of the PCM; f is the melting fraction of the PCM in element i.

Regeneration of the PCM during unoccupied hours is achieved through either admitting ambient air directly into the air channel or admitting the evaporatively cooled ambient air. The following equations represent the control strategy implemented to regenerate the PCM:

$$\begin{cases} if\ T_{amb} \leq T_{PCM,melting} - 1.5\text{°C} \quad then \quad Evaporative\ cooler\ OFF \\ \quad else \quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad Evaporative\ cooler\ ON \end{cases} \tag{6}$$

The direct single stage evaporative cooler was used and the model implemented to predict the temperature of the air leaving the evaporative cooler is the one adopted by Hourani et al. [25] and represented by the following equation:

$$T_{EC} = T_{amb} - PF(T_{amb} - T_{wb}) \tag{7}$$

Where T_{EC} is the temperature of the air leaving the evaporative cooler; PF is the performance factor of the evaporative cooler which could also be considered as its effectiveness and has a typical value in the range between 80 to 90% as reported by Hourani et al. [25] and experimented by Refaie et al. [27], so an average efficiency of 85% was used; T_{wb} is the wet bulb temperature of the ambient air.

Once the temperature of the last PCM element is below its melting temperature by 1°C then the PCM is completely regenerated and the evaporative cooler and the fan used to circulate the air will be turned off.

C. PEC and Thermal Comfort Model

The PEC extracts air from the room using a small fan and cools the air after passing it over a wet pad. This increases the moisture content of the air while decreasing its temperature. The PEC affects only the climate in the vicinity of the occupants, consequently affecting the head and the trunk of the occupant. To calculate the thermal comfort of the occupants in a non-uniform environment, Zhang et al. [28] developed a model capable of predicting the overall thermal comfort level using the segmental skin and core temperatures. To find the skin and core temperatures, Salloum et al. [29] developed a bioheat model capable of predicting the segmental temperature of 11 body segments (head, chest, back, abdomen, buttocks, upper arm, lower arm, hands, thighs, calves, and feet). Once the segmental temperatures are determined, the level of thermal comfort can be calculated and it is reflected by the scale, developed by Zhang et al. [28], ranging from -4 (very uncomfortable) to +4 (very comfortable).

The different types of evaporative cooling pads including corrugated paper, aspen fiber, and rigid cellulose have a high average performance factor of 90% as experimented by

Kulkarni et al. [30]. To estimate the temperature of the air leaving the PEC nozzle, the same model used for the evaporative cooler can be used.

D. Desiccant Wheel Model

The type of desiccant used in this study is Silica gel, due to its high absorption capacity which can go up to 40% of its weight [31]. Moreover, its adsorption characteristics over a wide range of humidity makes it the first choice for solid desiccants [31]. The psychometric model developed and experimentally validated by Beccali et al. [31] and used by Hourani et al. [25] was employed to predict the performance of the solid desiccant wheel of the proposed system. The model was reported to be valid if the supply and regeneration air were in a counter flow arrangement with equal flow rates [32].

The model predicts the conditions of the dehumidified air (enthalpy and relative humidity) given the conditions of the outdoor and the regeneration air, and the equations are shown below:

$$H_{out} = 0.1312\, H_{reg} + 0.8688\, H_{amb} \tag{8}$$

$$\emptyset_{out} = 0.9428\, \emptyset_{reg} + 0.0572\, \emptyset_{amb} \tag{9}$$

Where H_{out}, H_{reg}, and H_{amb} are the enthalpies of the dehumidified air leaving the desiccant wheel, the regeneration air and the ambient air (kJ/kg), respectively; ϕ_{out}, ϕ_{reg}, and ϕ_{amb} are the relative humidity of the dehumidified air leaving the desiccant wheel, the regeneration air and the ambient air (%), respectively.

The model correlations were reported to be valid for inlet temperatures between 20 °C and 34 °C, regeneration temperatures between 40 °C and 80 °C, inlet air humidity ratios between 8 and 15 g/kg, and regeneration air humidity ratios between 10 and 16 g/kg [31].

E. Regeneration of the Desiccant Wheel

A heat source is needed in order to regenerate the desiccant wheel. This heat source can be obtained from the free energy provided by the sun during the day. Thus, the use of the Trombe wall to heat the air for regenerating the desiccant made it a passive sustainable strategy to be employed. The Trombe wall heats the ambient air flowing through the channel between the wall and the glazing, and then the heated air passes through the desiccant wheel for regeneration. A similar configuration of the Trombe wall as Ong et al. [33] and Badawiyeh et al. [34] was employed in this study such that 1D quasi-steady heat transfer through the Trombe wall was assumed while considering uniform temperatures within each of the glazing and the air channel. The flow rate passing through the air channel of the Trombe was set equal to the flow rate of the fresh air passing through the desiccant wheel. The model takes the ambient weather data, glazing properties, wall properties, dimensions of the air channel and the air flow passing through it as the inputs (more details can be found in section 5). The outputs of the model are the temperatures of the glazing and air channel (uniform temperatures), and the external boundary of the wall.

The energy balance equations of the glazing, air channel, and wall are used to obtain the outputs of the model and are shown below respectively:

$$\alpha_g I + h_{rad}(T_W - T_g) + h_{ch-g}(T_{ch} - T_g) = h_{amb}(T_g - T_{amb}) + Q_{rad-amb} \quad (10)$$

$$h_{ch-W}(T_W - T_{ch}) = h_{ch-g}(T_{ch} - T_g) + \frac{\dot{m}_{ch} C_p (T_{ch} - T_{amb})}{A_{ch}} \quad (11)$$

$$\tau\alpha_W I = h_{ch-W}(T_W - T_{ch}) + h_{rad}\left(T_W - T_g\right) + Q_{cond} \tag{12}$$

Where the first, second and last terms of the left side of Eq. (10) represent respectively the absorbed solar radiation by the glazing, the radiation between the wall and the glazing, and the convection between the air flowing through the channel with the glazing. The first and last terms of the right side correspond respectively to the convection between the ambient air and the glazing and the radiation between the glazing and the ambient; the last term of the right side of Eq. (11) represents the flow of air through the channel; the first term of the left side of Eq. (12) is the portion of solar radiation transmitted through the glazing and absorbed by the wall; Q_{cond} is the heat conducted through the wall. A schematic of the Trombe wall along with the parameters used are illustrated in **Fig.3**.

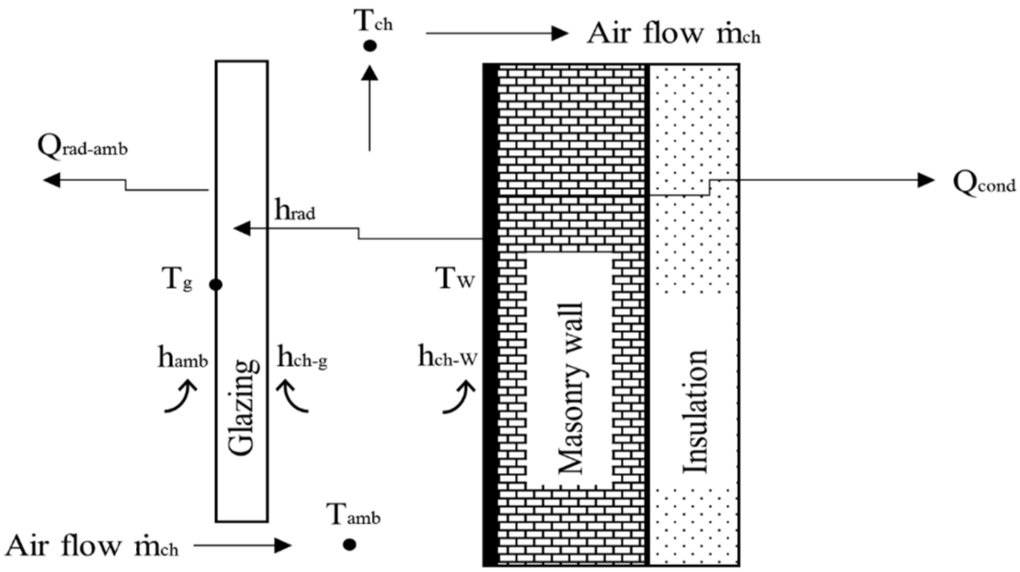


Fig. 3: Schematic of the Trombe wall

The air leaving the channel of the Trombe wall at an elevated temperature (T_{ch}) is used for regenerating the desiccant wheel. If the Trombe wall was not enough to heat the air to the required regeneration temperature, other means of heating were implemented such as auxiliary heating.

CHAPTER IV

NUMERICAL METHODOLOGY

The sequence of operation of the proposed system is summarized by the flow chart shown in **Fig. 4**. The input parameters of the system were first defined at each hour of operation. After that, the desiccant wheel model was implemented to find the characteristics of the dehumidified air using the ambient conditions along with the regeneration conditions. Next, the PCM model was used to find the supply temperature using Eq. (3) which represented 1D transient heat transfer between the PCM layer and the air channel, where the PCM layer was made up of PCM bricks that are 10 cm long, and the air channel was discretized accordingly into elements of equal length of 10 cm. After that, the space model was used to calculate the room macroclimate air temperature and humidity using Eq. (1) and Eq. (2), respectively, where each envelope component (walls, ceiling, and floor) was discretized into 24 elements. Then, the PEC model was used to calculate the temperature of the air reaching the occupant's upper body parts so that the thermal comfort model was used to assess the thermal comfort of the occupants. The equations were solved using the finite volume method and an implicit scheme was implemented. The use of an implicit scheme allowed for a larger time step to be adopted, so a time step of 360 s was used to reduce the simulation runtime. The PCM, space and PEC models were coupled, so the output parameters were initialized and were obtained by iterations until convergence was reached with an error less than 10^{-4}. After convergence was attained, the model moved to the next time step until the simulation time ends. The simulation was performed for several days to eliminate the effect of initialization on the performance of the system and to obtain a steady periodic solution.

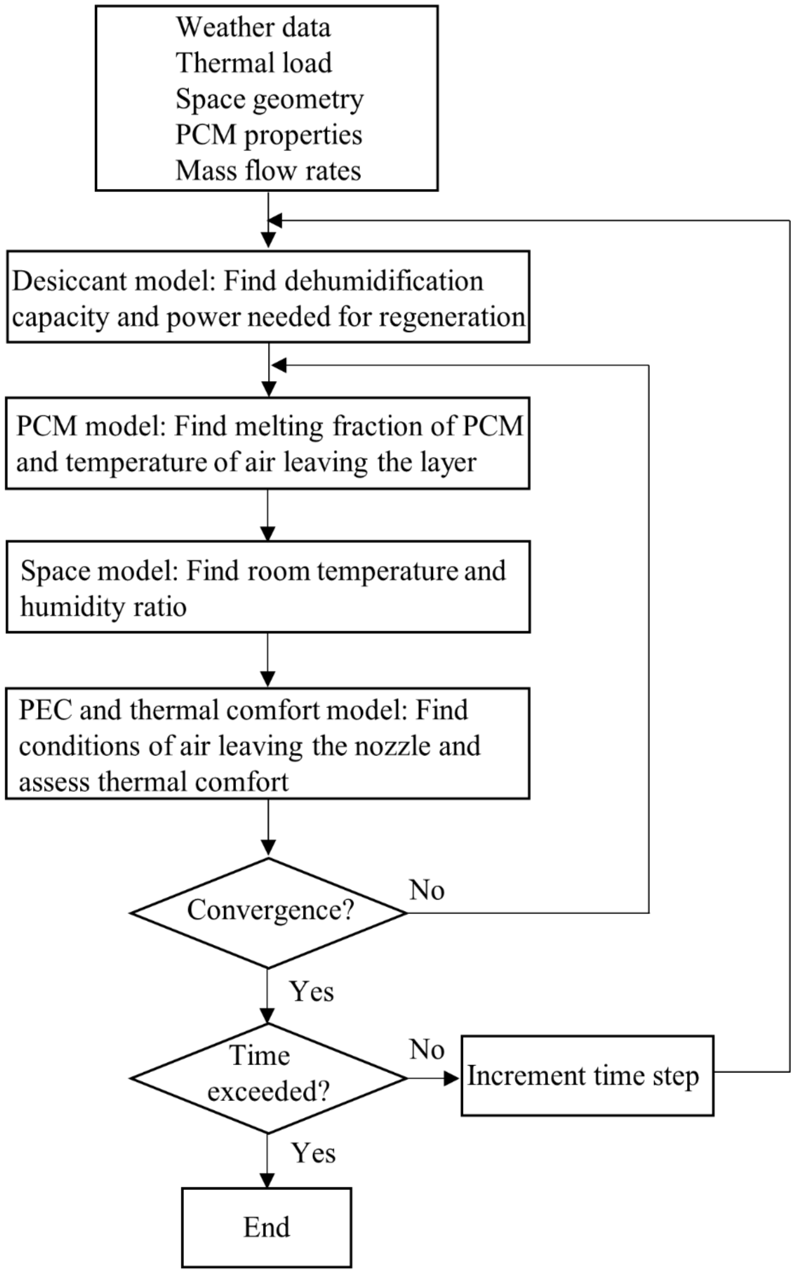


Fig. 4: Flow chart of the proposed system

CHAPTER V

CASE STUDY

The current study evaluated the performance of the proposed system in providing indoor thermal comfort. A case study of an office space (7 m × 6 m × 3 m) in Beirut was simulated during the summer months from June to September, in which a typical representative day for each month was simulated. The weather data was obtained using TMY (Typical Meteorological Year) for Beirut using Meteonorm (V7.3) database. The TMY produces typical weather data as it uses statistics performed over a number of years to present data of the highest occurrence. For Meteonorm, the TMY data is predicted based on a 10 year period extending from 2000 to 2009, and the data for a representative day for each month is shown in **Table 1.**

Table 1: Ambient conditions for typical days in each summer month

	June			July			August			September		
Time	T_{amb} (°C)	T_{wb} (°C)	I (W/m^2)	T_{amb} (°C)	T_{wb} (°C)	I (W/m^2)	T_{amb} (°C)	T_{wb} (°C)	I (W/m^2)	T_{amb} (°C)	T_{wb} (°C)	I (W/m^2)
00:00	23	19	0	27	22	0	27	22	0	26	21	0
1:00	23	18	0	27	22	0	27	21	0	26	21	0
2:00	23	18	0	27	21	0	27	21	0	26	20	0
3:00	23	18	0	27	21	0	27	20	0	26	20	0
4:00	23	18	0	26	21	0	27	20	0	25	20	0
5:00	23	18	0	26	21	0	27	20	0	25	20	0
6:00	23	18	14	26	21	3	26	21	0	25	20	0
7:00	23	19	270	26	21	218	27	21	126	25	20	44
8:00	25	19	445	28	22	420	28	22	443	26	21	410
9:00	25	20	525	28	22	511	29	22	577	27	21	585
10:00	26	21	567	29	22	558	29	23	640	28	21	660
11:00	26	21	590	29	23	583	30	23	673	28	22	697

12:00	26	22	602	30	23	596	30	24	690	28	22	715
13:00	27	22	607	30	24	601	30	24	694	29	22	721
14:00	27	23	604	30	24	599	30	25	681	29	22	716
15:00	27	23	594	30	24	590	30	25	655	29	23	700
16:00	27	23	574	30	24	570	30	25	607	29	23	665
17:00	26	23	538	29	24	535	30	25	510	28	23	595
18:00	26	23	471	29	24	468	29	25	285	28	23	436
19:00	26	23	331	28	24	328	29	25	6	28	23	67
20:00	26	22	49	27	24	48	28	25	0	28	23	0
21:00	26	22	0	27	24	0	28	24	0	27	22	0
22:00	25	21	0	27	23	0	28	24	0	27	22	0
23:00	25	20	0	27	23	0	28	23	0	27	21	0

The office was occupied from 8:00am to 5:00pm which is the typical working schedule in Lebanon. The occupancy schedule, shown in **Fig. 5,** was obtained such that the maximum number of occupants present in the office space at once was 3 following the recommendation of ASHRAE [6] with a minimum of 14 m^2 of office area per occupant. According to ASHRAE [6], each occupant contributes 75 W sensible load and 55 W latent load to the space load. The lighting and equipment loads were estimated at 84 W per occupant [6].

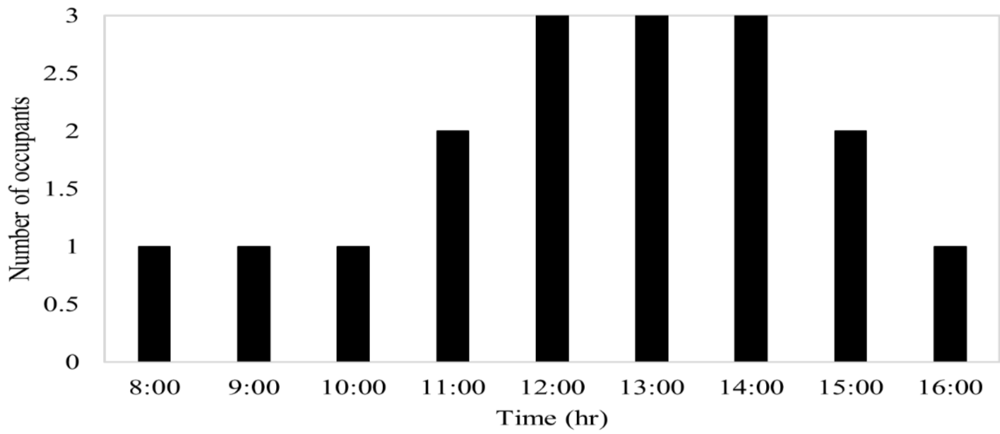


Fig. 5: Occupancy schedule in the space

The Trombe wall was incorporated into the building envelope at the south wall which was made up of a clear single glazing, an air channel (4 cm thick) as recommended by Kessel [35], and a double layered wall composed of a polystyrene insulation layer and a masonry concrete wall coated with a dark heat absorbing material as used by Badawiyeh et al. [34]. The space was also made up of an external wall on the east side while the remaining space envelope were internal surfaces partitioned with conditioned spaces, and their properties were obtained from the thermal standards for buildings in Lebanon [36]. The properties of the Trombe wall and the space envelope properties are found in **Table 2**.

Table 2: Space envelope properties

Parameter	Properties
Glazing	Clear single glazing; α = 0.06; ε = 0.9; τ = 0.84
South wall	5 cm polystyrene insulation layer; k = 0.032 W/m·K; α = 0.95; ε = 0.95; ρ = 70 kg/m^3; C_p = 830 J/kg·K 25 cm masonry concrete, k = 1.8 W/m·K; ρ = 2800 Kg/m^3; C_p = 1200 J/kg·K
East wall	20 cm concrete Hollow block ; U = 3.03 W/m^2·K; ρ = 600 kg/m^3; C_p = 1200 J/kg·K
Internal envelope	10 cm concrete Hollow block ; U = 3.63 W/m^2·K; ρ = 600 kg/m^3; C_p = 1200 J/kg·K

The supply of fresh air was chosen based on ASHRAE standards [6] of minimum required ventilation rate per occupant (7 *l*/s per person), and the flow rate of the air passing through the Trombe wall was equal to the supply of fresh air as recommended by Beccali et al. [32]. The PEC flow rate was selected based on human thermal comfort standards (7 *l*/s) [20], and it was maintained constant throughout the occupancy hours and throughout the simulations for all of the months. The supply flow rate of air to the space was chosen based on ASHRAE minimum ventilation rate (92 *l*/s) [6]. A control strategy was adopted to supply the outdoor air to the space 3 hours prior to the occupancy schedule to refresh the room air with the cooler and less humid ambient air with a flow rate equal to the supply flow rate during occupied hours (92 *l*/s). For the other unoccupied hours, a small flow rate of outdoor air was infiltrated to the space (2 *l*/s). The flow of the air used to regenerate the PCM during nighttime was 11.5 *l*/s. It was selected such that a complete regeneration was obtained just before the occupancy schedule at the month that had the peak ambient conditions for temperature and humidity. For Beirut climate, it was the month of August.

A. PCM Selection

The PCM was stored in a thin layer above the insulated ceiling extending to nearly the entire area of the roof. To select the PCM melting temperature and mass, simulations were done for the month with the peak conditions. In this case, the highest outdoor air temperature and the most humid weather throughout daytime and nighttime were in August. The PCM melting temperature was selected such that the room macroclimate temperature does not exceed the maximum limit for thermal comfort which is 29 °C while using a PEC to provide thermal comfort to the occupants [16]. However, to completely regenerate the PCM during unoccupied hours, even with the aid of evaporative cooling, the minimum melting PCM temperature was 25 °C. To sustain comfort for the entire occupied hours, a minimum of 7.87 kg (5.5m × 6.5m with a thickness of 0.25 mm) of PCM should be present. The PCM properties were obtained from Thambidurai et al. [37] and are listed in **Table 3**.

Table 3: PCM properties

PCM property	
Specific heat	2000 J/kg·K
Density	880 kg/m^3
Latent heat of fusion	210/kg

B. Regeneration Temperature Selection

The regeneration temperature of a desiccant wheel using Silica gel as the desiccant should be within the range from 40 °C to 80 °C as recommended by Beccali et al. [31]. The regeneration temperature was fixed throughout the day for each month while maintaining a room relative humidity below 65 % which is the maximum allowable limit to preserve thermal comfort as set by ASHRAE [6]. Simulations were performed for June, July, August, and September, and the minimum regeneration temperature to keep the relative humidity of the space just below the maximum allowable limit was found to be 55 °C, 51 °C, 57 °C, and 40 °C, respectively. Whenever the Trombe wall was not enough to obtain the required regeneration temperature, the auxiliary heater was utilized to provide the desired regeneration throughout the day.

CHAPTER VI

RESULTS AND DISCUSSION

The proposed system is assessed based on the simulations of the developed mathematical model, and the level of the indoor thermal comfort and the energy consumption for each summer month was evaluated. The effectiveness of the Trombe wall in reducing energy consumption was assessed by comparing the energy consumption of the system with and without the Trombe wall. Finally, the system performance was evaluated by finding the energy savings the system can yield compared with a conventional AC unit. The system was simulated for a typical day for each summer month (June, July, August, and September) with August being the most critical month.

Fig. 6 to **Fig. 9** present the variation in time of (a) the temperature of the room and ambient air while showing the temperature range of the PEC outlet and the overall comfort levels during occupied hours and (b) the humidity ratio of room and ambient air for the months of June, July, August and September, respectively. The desiccant wheel, Trombe wall and PEC were turned on during the occupied hours, while they were off during the unoccupied hours. The flow rate of air passing through the desiccant wheel and the Trombe wall varies per the occupancy schedule, while the PEC flow rate was maintained constant throughout the occupancy hours.

For the month of June (**Fig. 6**), the room temperature varies between 24.8 °C and 26.3 °C during occupancy hours (see **Fig. 6 (a)**) which is below the maximum allowable room temperature of 29 °C. The room temperature was observed to increase steadily as the occupancy hours begin while decreasing slightly towards the end of the occupancy schedule which is due to the change of the occupancy schedule and thus a decrease in the internal load. During unoccupied hours, the room temperature slightly decreases with the decrease of the ambient through exchanging heat with the walls and a small flow rate of ambient air due to infiltration. Before 3 hours of the occupancy schedule, the supply fan is turned on to refresh the ambient air which is cooler and less humid than the room air, hence a noticeable decrease in the temperature and humidity of the room air. The PEC outlet temperature varies slightly between 19.4 °C and 21.5 °C (see **Fig. 6 (a)**). The overall thermal comfort of the occupants was found using Zhang's thermal comfort model [28] and it was between -0.34 and +0.32, which falls above the limit of the minimum acceptable thermal comfort of -1 or slightly uncomfortable. Even though the PEC increased the moisture generation in the space, desiccant dehumidification maintained the room relative humidity between 58% and 63% below the maximum allowable limit of 65%.

As for the months of July, August, and September (**Fig. 7** to **Fig. 9**), they had similar weather conditions with August being the hottest and most humid month. The room temperature remained below the maximum allowable macroclimate temperature of 29 °C where it reached a maximum of 28.2 °C in August (see **Fig. 8(a)**). In addition, with the aid of the desiccant wheel the room relative humidity remained below 65 % during the occupied hours while varying the minimum regeneration temperature for each month. The overall comfort of the occupants was also just at the limit of the minimum thermal comfort level while obtaining slightly better results for July and September, which have a drier cooler climate than August.

For September, the humidity ratio of the room has a different trend than the other months (see **Fig. 9(b)**) where the room humidity ratio is higher than the ambient in the first hours of occupancy. This occurs due to the relatively low ambient humidity and high temperature such that even with the moisture generation from the PEC and occupants in the space, the room relative humidity remains below 65% and comfort is attained. As for the remaining occupancy hours towards the end of the day, the ambient humidity was high, and the moisture generation increased with the increase of the number of occupants as per the occupancy schedule. Hence, dehumidification was needed to maintain the thermal comfort limit for the maximum allowable humidity in the space. Hence, the desiccant dehumidification increases to offset of the humidity in the space. In addition, the Trombe wall provided extra regeneration above the minimum required limit. Therefore, the relative humidity in the space during the last hours of occupancy was in the range between 58% and 62% which fell well below the maximum allowable limit of 65%.

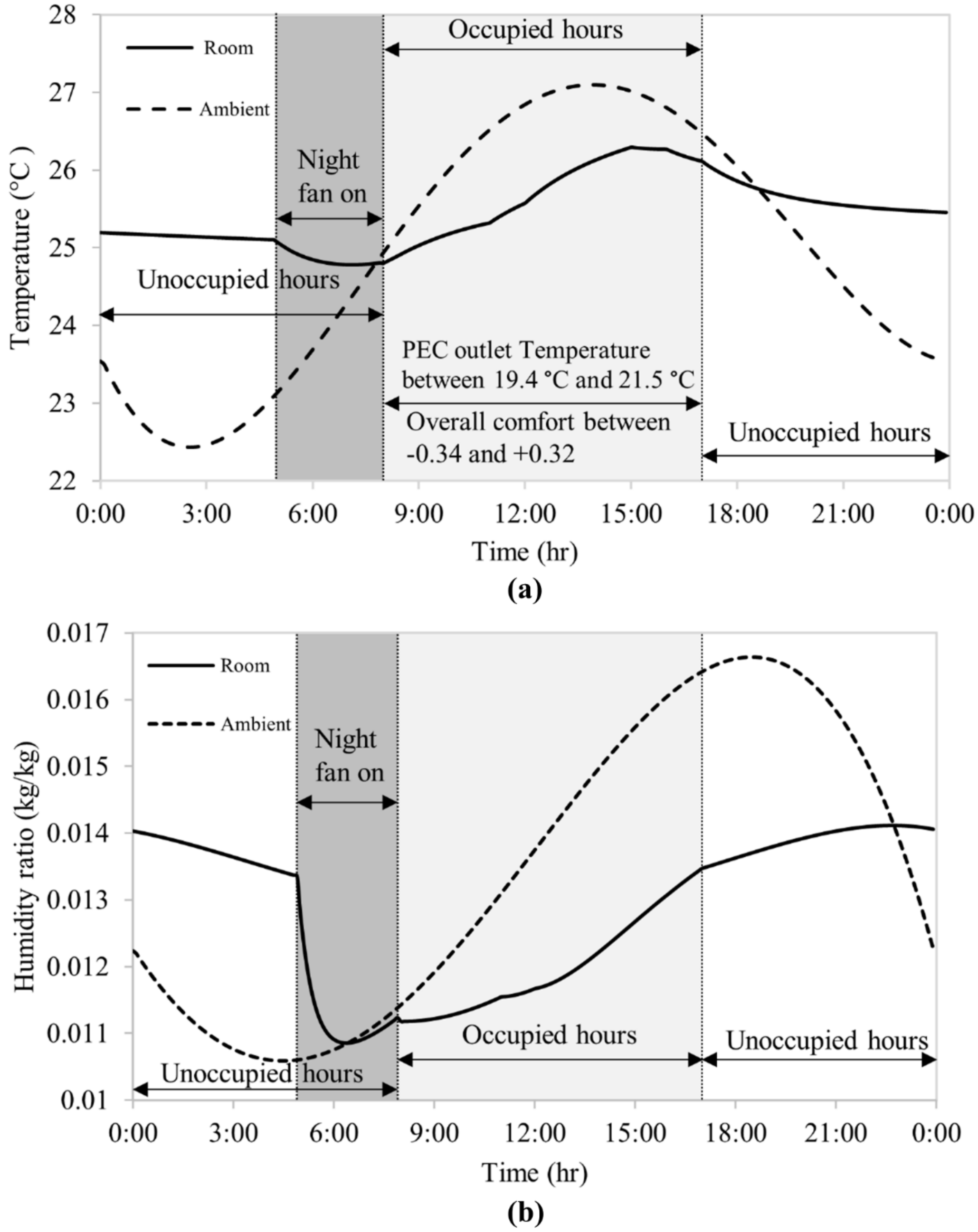
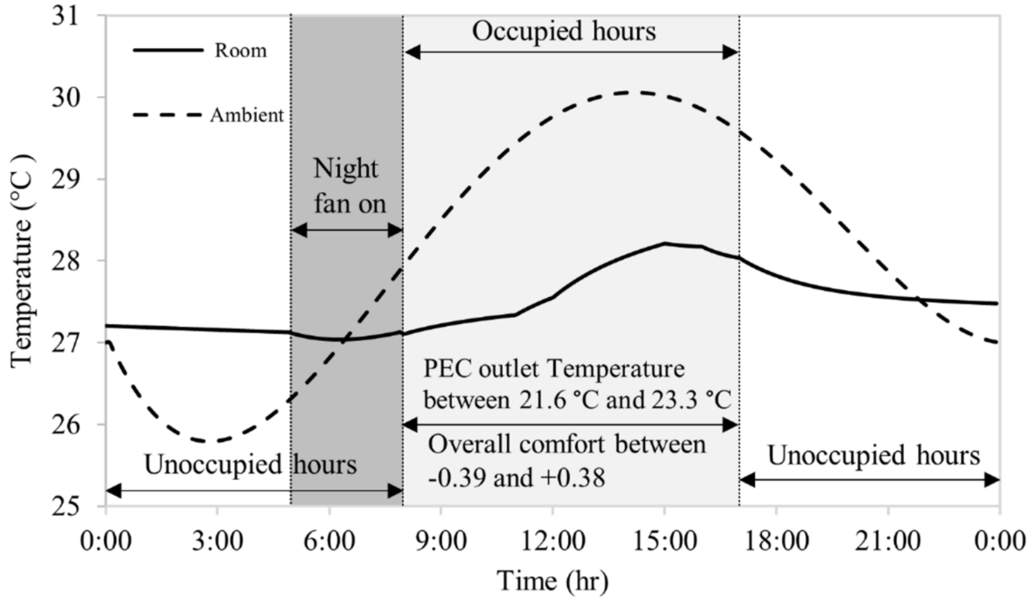
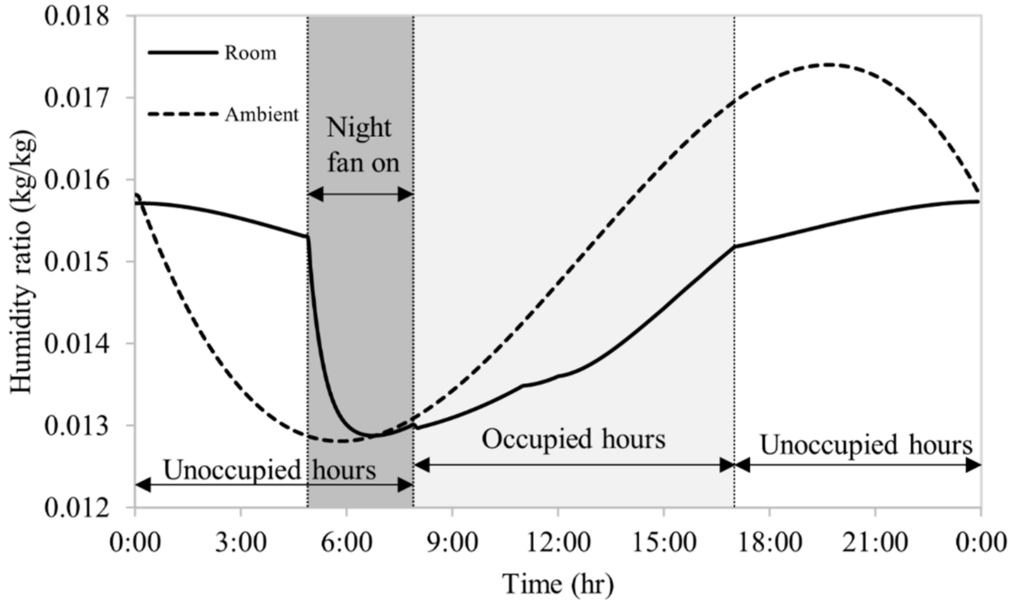


Fig. 6: The variation in time of (a) the temperature of the room and ambient air while showing the temperature range of the PEC outlet and the overall comfort levels during occupied hours and (b) the humidity ratio of room and ambient air for the month of June



(a)



(b)

Fig. 7: The variation in time of (a) the temperature of the room and ambient air while showing the temperature range of the PEC outlet and the overall comfort levels during occupied hours and (b) the humidity ratio of room and ambient air for the month of July

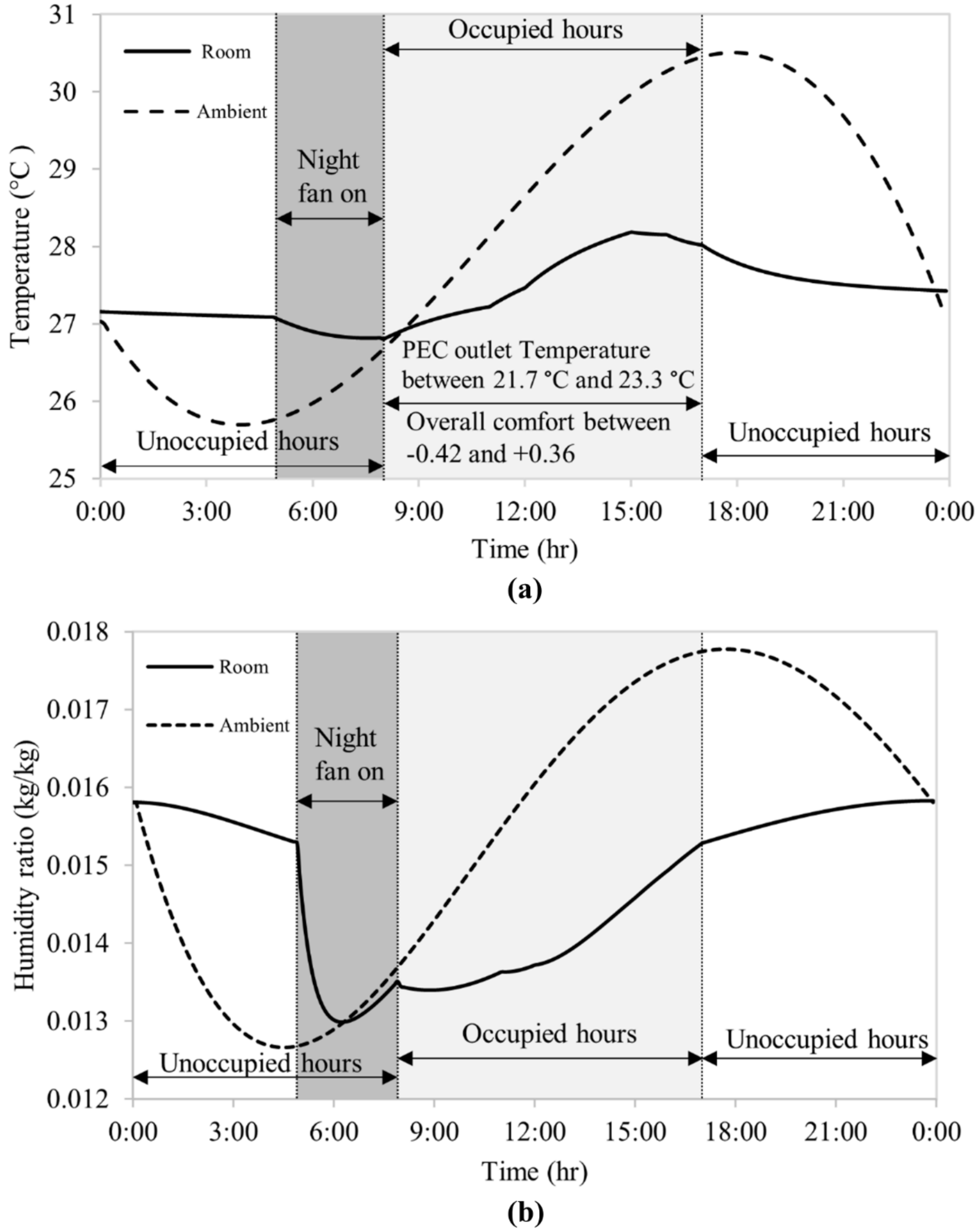
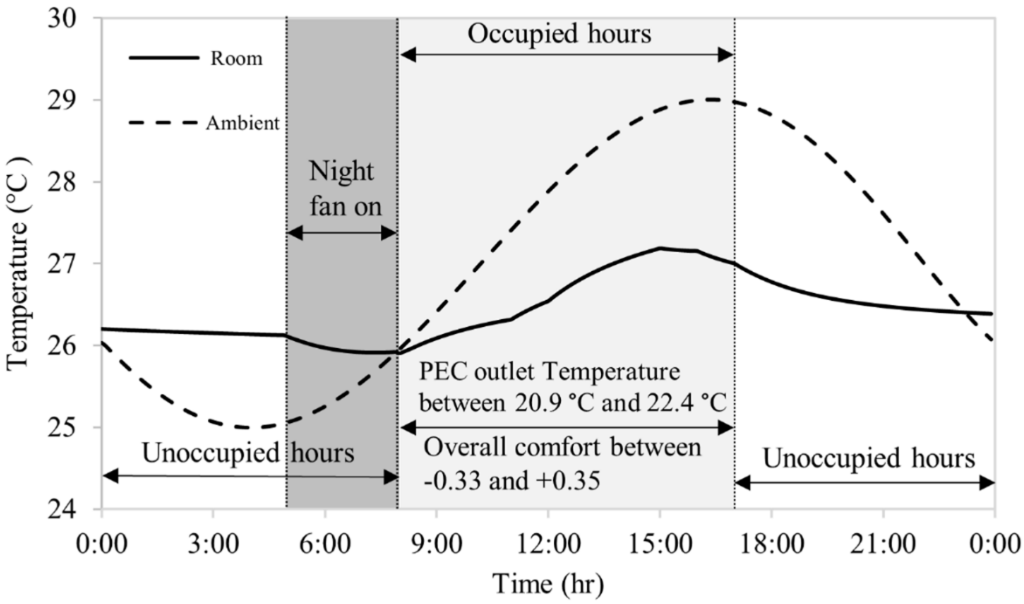
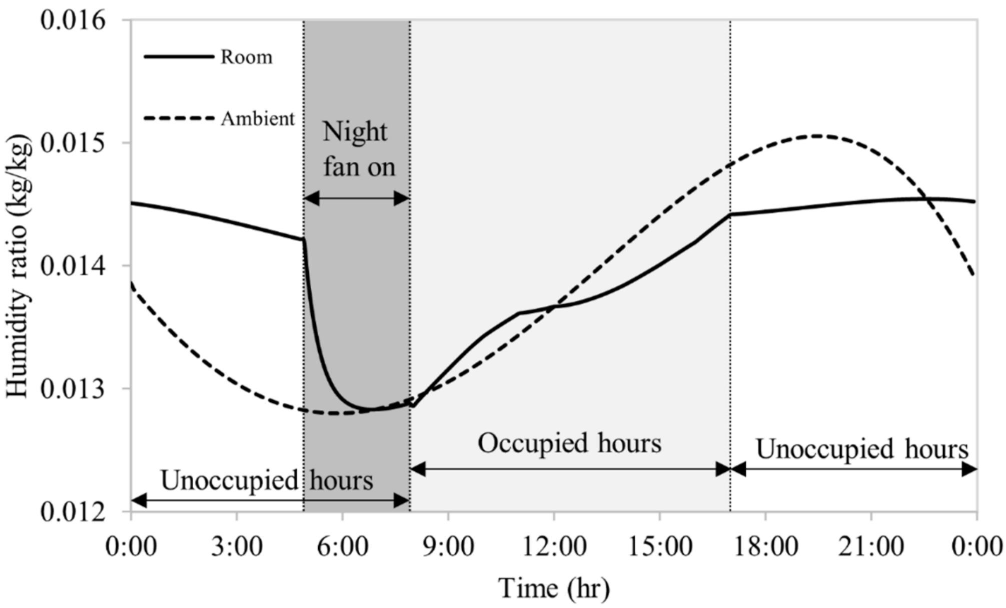


Fig. 8: The variation in time of (a) the temperature of the room and ambient air while showing the temperature range of the PEC outlet and the overall comfort levels during occupied hours and (b) the humidity ratio of room and ambient air for the month of August



(a)



(b)

Fig. 9: The variation in time of (a) the temperature of the room and ambient air while showing the temperature range of the PEC outlet and the overall comfort levels during occupied hours and (b) the humidity ratio of room and ambient air for the month of September

As for the PCM, it completely regenerated during the night in all months with the aid of evaporative cooling of the ambient air except for the month of June, where the ambient air was cool enough for regeneration and the evaporative cooler was unnecessary. The flow rate of the air used for regeneration was the minimum flow rate to obtain complete regeneration just before the occupancy hours begin for the most critical month (August). The PCM completely melted only during August, and it melted at the end of the occupied hours lasting the whole duration of the occupancy (9 hours). Regeneration occurred faster in the months other than August as the PCM did not completely melt. **Table 4 summarizes t**he melting fraction at the end of the occupancy hours for each month, and the duration needed for complete regeneration with the number of hours when the evaporative cooler.

Table 4: Melting fraction at the end of the occupancy hours and the duration needed for complete regeneration and number of hours when evaporative cooling was used

	Melting fraction at the end of the occupancy hours	**Duration of complete regeneration in hours**	**Number of hours when the evaporative cooler was on**
June	0.26	3.1	0.0
July	0.93	7.8	7.8
August	1.00	8.3	8.3
September	0.53	5.7	5.7

If the heat from Trombe wall was not enough to heat the air to the minimum needed regeneration temperature for each month, the auxiliary heater was used. **Table 5** presents the hourly energy consumption of the auxiliary heater (Q_{aux}) and the temperature of the air leaving the Trombe wall channel ($T_{channel}$). The temperature of the air leaving the Trombe wall varied hourly due to the time-variation of the solar irradiance and ambient temperature. The energy consumption of the auxiliary heater also varied hourly as per the occupancy schedule as more energy was needed to heat a larger flow of air. The consumption of the heater depended on the variation of the temperature of the air leaving the Trombe wall channel. It was observed that for the month of September, which had highest solar irradiance compared to other months, the Trombe wall alone was nearly enough to obtain a room relative humidity below the maximum limit of 65 %. However, in June which had the lowest solar irradiance and had a relatively high humidity, the total auxiliary heater consumption was found to be the highest among the different months.

Table 5: Temperature of the air leaving the Trombe wall and the needed auxiliary heater power for regeneration during each month

Month	June (T_{reg} = 55 °C)			July (T_{reg} = 51 °C)			August (T_{reg} = 57°C)			Sep (T_{reg} = 40 °C)		
Time	T_{amb} (°C)	T_{ch} (°C)	Q_{aux} (W)	T_{amb} (°C)	T_{ch} (°C)	Q_{aux} (W)	T_{amb} (°C)	T_{ch} (°C)	Q_{aux} (W)	T_{amb} (°C)	T_{ch} (°C)	Q_{aux} (W)
8:00 – 9:00	25	34	176	28	37	122	28	35	187	26	34	55
9:00 – 10:00	25	37	151	28	40	97	29	40	147	27	39	10
10:00 – 11:00	26	38	142	29	41	85	29	42	130	28	42	0
11:00 – 12:00	26	38	273	29	41	164	30	42	245	28	43	0
12:00 – 13:00	26	38	420	30	41	253	30	42	383	28	43	0
13:00 – 14:00	27	38	425	30	41	255	30	42	382	29	43	0
14:00 – 15:00	27	38	423	30	41	252	30	43	365	29	44	0
15:00 – 16:00	27	39	284	30	42	160	30	44	232	29	45	0
16:00 – 17:00	27	40	138	30	43	73	30	46	107	29	47	0

The use of Trombe wall for regeneration purposes has not been reported in literature. Hence, it is of interest to assess its effectiveness in reducing the energy consumption for the simulated case study for each summer month. To find the energy needed for regenerating the desiccant wheel, the heat energy needed to heat the ambient air to the required minimum regeneration temperature was calculated. The auxiliary heater energy consumption and the Trombe wall energy savings are shown in **Fig. 10**. The Trombe wall significantly reduced the thermal energy consumption especially in September since it had the highest solar irradiance on the south wall. In September, the auxiliary heater consumed only 1.9 kWh which was nearly 36 times less than the consumption of the system relying only on the auxiliary heater for regeneration. Over the entire summer period, the Trombe wall reduced the total thermal energy consumption by 55 % compared with using the auxiliary heater solely for regeneration.

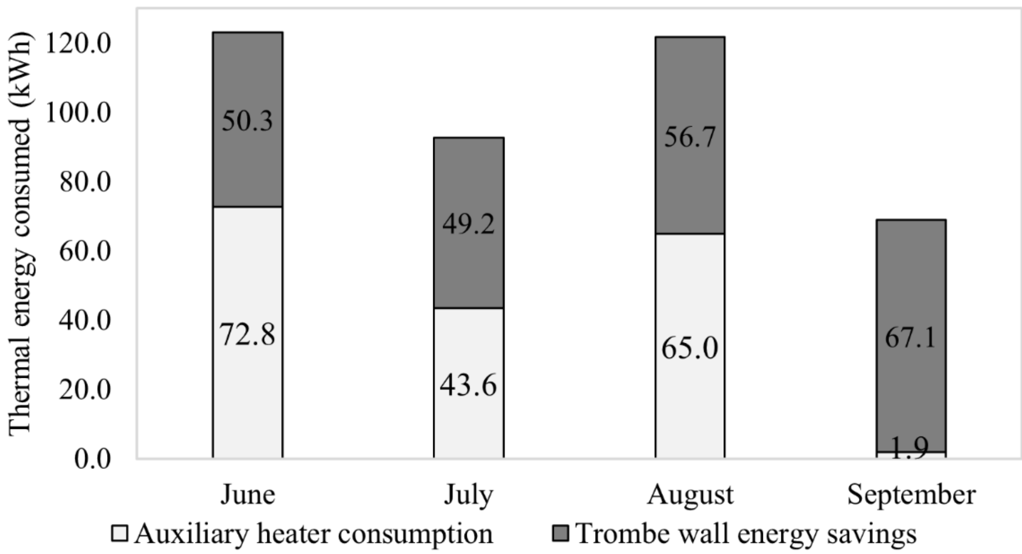


Fig. 10: Thermal energy consumption of the auxiliary heater of the proposed system and the energy savings of the Trombe wall

The performance of the proposed cooling system was evaluated by comparing its consumption with the consumption of a mechanical ventilation system that utilizes a direct expansion system with a typical coefficient of performance of 3.0 [34] to obtain the same comfort level. The average thermal comfort obtained using the proposed system on Zhang et al. [28] scale is -0.010, -0.005, -0.030, 0.010 for June, July, August, and September, respectively, which correspond to a nearly neutral state on the PMV scale. Simulations were performed for each month to find the electric energy needed to remove the sensible and latent loads for the same room envelope, supply air flow rate, fresh air flow rate, and ambient conditions used for the proposed system.

As for the electric consumption of the proposed system, it was made up of the consumption of the fans of the supply air, fresh air, air used to regenerate the PCM, air passing through the Trombe wall, air passing through the sensible wheel, and the PEC as well as the power to rotate the desiccant wheel. Comparing the performance of the proposed system with a conventional AC unit, **Table 6** presents the electrical energy consumption of both the proposed system and the conventional one. The results showed significant electrical savings especially in August where the electrical energy saving was 159.4 kWh which amount to 96% energy reduction. The overall electrical energy saving for the entire summer period was 513.5 kWh corresponding to nearly 95% of the electric consumption of the conventional AC system.

Table 6: Electrical energy consumed by a conventional AC unit compared with the proposed system during the entire month

Electrical consumption (kWh)	Conventional AC unit	Proposed system

June	95.6	6.0
July	158.3	6.2
August	165.7	6.3
September	118.5	6.1

The cost of operation of each system per month was calculated and the results are shown in **Fig. 11.** The calculations were based on a pricing of 5 cents/ kWh of central heating and 20 cents/ kWh of electrical energy. The proposed system achieved significant savings especially in the month of September where the conventional system costs $23.7 which was nearly 18 times more than the proposed system. Even in the month of June, the savings remained substantial as the system saved up to 75 % compared with the conventional AC system. Overall, the proposed passive system achieved savings amounting to 87% over the entire summer period from the conventional AC system.

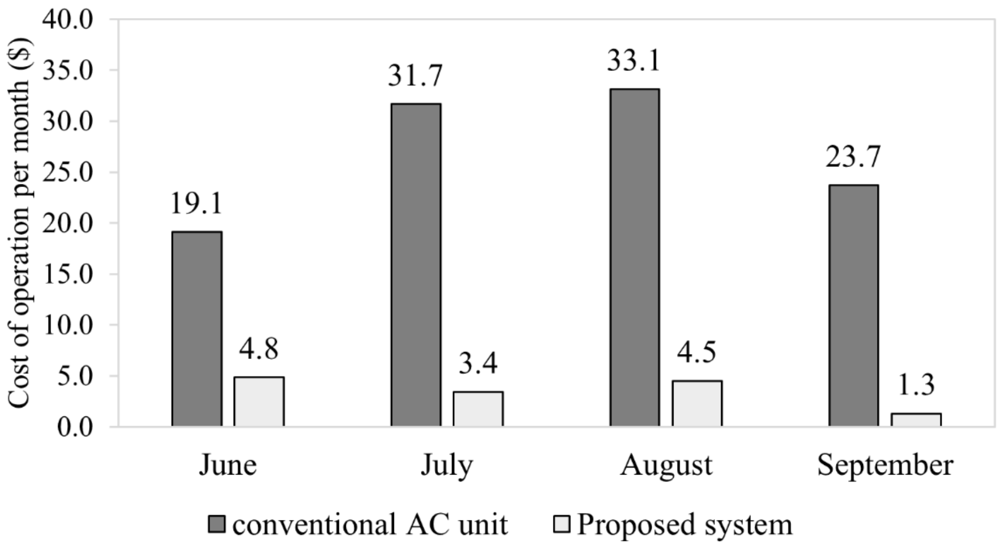


Fig. 11: The cost of operating a conventional AC unit and the proposed system for each month

CHAPTER VII

CONCLUSION

This paper evaluated the integration of a hybrid cooling system in an office space made up of a PCM layer, a personalized evaporative cooler, and a solid desiccant wheel regenerated using an auxiliary heater aided with a Trombe wall. The system proved its feasibility as an 87% reduction of the total operating cost was achieved compared with a conventional cooling system over the entire cooling season. The novelty of the proposed system stems from using Trombe wall for regeneration. Accordingly the Trombe wall savings were evaluated and it was found that the Trombe wall can save up to 55% of the thermal energy consumption of the proposed system over the entire summer period compared with having to rely only on the auxiliary heater for regeneration. It should be noted that another source of heating could be implemented simultaneously with the Trombe wall instead of the auxiliary heater, such as solar energy or a waste heat source, to improve the performance of the system in terms of energy savings.

BIBLIOGRAPHY

[1] R. Zeinelabdein, S. Omer, G. Gan, Critical review of latent heat storage systems for free cooling in buildings, Renewable and Sustainable Energy Reviews 82 (2018) 2843-2868.

[2] L. Yang, Y. Haiyan, J. C. Lam, Thermal comfort and building energy consumption implications – A review, Applied Energy 115 (2014) 164-173.

[3] E. Cuce, D. Harjunowibowo, P. M. Cuce, Renewable and sustainable energy saving strategies for greenhouse systems: A comprehensive review, Renewable and Sustainable Energy Reviews 64 (2016) 34-59.

[4] R. Zeinelabdein, O.S., E. Mohamed, I. Amaireh, G. Gan, Free cooling based change material for domestic buildings in hot arid climate, in 16th International Conference on Sustainable Energy Technologies, Bologna, Italy, 2017.

[5] D. Lefebvre, H. F. Tezel, A review of energy storage technologies with a focus on adsorption thermal energy storage processes for heating applications, Renewable and Sustainable Energy Reviews 67 (2017) 116-125.

[6] ASHRAE, Thermal Environmental Conditions For Human Occupancy, 2013.

[7] M. Dardir, K. Panchabikesan, F. Haghighat, M. El Mankibi, Y. Yuan, Opportunities and challenges of PCM-to-air heat exchangers (PAHXs) forbuilding

free cooling applications—A comprehensive review, Journal of Energy Storage 22 (2019) 157-175.

[8] A. Waqas, Z. Ud Din, Phase change material (PCM) storage for free cooling of buildings—A review, Renewable and Sustainable Energy Reviews 18 (2013) 607-625.

[9] M. Anisura, M. Mahfuza, M. Kibriaa, R. Saidura, I. Metselaara, T. Mahlia, Curbing global warming with phase change materials for energy storage, Renewable and Sustainable Energy Reviews 18 (2013) 23-30.

[10] P. Tatsidjodoun, N. Le Pierrès, L. Luo, A review of potential materials for thermal energy storage in building applications, Renewable and Sustainable Energy Reviews 18 (2013) 327-349.

[11] D. Fernandes, F. Pitié, G. Cáceres, J. Baeyens, Thermal energy storage: "How previous findings determine current research priorities", Energy 39 (2012) 246-257.

[12] F. Souayfane, F. Fardoun, P.-H. Biwole, Phase change materials (PCM) for cooling applications in buildings: A review, Energy and Buildings 129 (2016) 396-431.

[13] E. Osterman, V. Butala, U. Stritih, PCM thermal storage system for 'free' heating and cooling of buildings, Energy and Buildings 106 (2015) 125-133.

[14] Y. Zhang, R. Zhao, Effect of local exposure on human responses, Building and Environment 42 (2007) 2737–2745.

[15] G. Knudsen, A. Melikov, Human response to individually controlled environment, in Proceedings: Indoor Air, Lyngby, Denmark, 2005.

[16] H. Zhang, Human thermal sensation and comfort in transient and non-uniform thermal environments, University of California, Berkeley: PhD thesis, 2003.

[17] R. Cermak, A. Melikov, L. Forejt, O. Kovar, Performance of personalized ventilation in conjunction with mixing and displacement ventilation, HVAC&R Research 12 (2006) 295-311.

[18] S. Sekhar, N. Gong, K. Tham, K. Cheong, A. Melikov, D. Wyon, P. Fanger, Findings of personalized ventilation studies in a hot and humid climate, HVAC & R Research 11 (2005) 603-620.

[19] J. Kaczmarczyk, A. Melikov, P. Fanger, Human response to personalized ventilation and mixing ventilation, Indoor Air 14 (2004) 18-29.

[20] W. Chakroun, N. Ghaddar, K. Ghali, Chilled ceiling and displacement ventilation aided with personalized evaporative cooler, Energy and Buildings 43 (2011) 3250-3257.

[21] S. Yamaguchi, K. Saito, Numerical and experimental performance analysis of rotary desiccant wheels, International Journal of Heat and Mass Transfer 60 (2013) 51-60.

[22] P. Bevilacqua, F. Benevento, R. Bruno, N. Arcuri, Are Trombe walls suitable passive systems for the reduction of theyearly building energy requirements?, Energy 185 (2019) 554-566.

[23] X. Hong, M. K. Leung, W. He, Effective use of venetian blind in Trombe wall for solar space conditioning control, Applied Energy 250 (2019) 452- 460.

[24] B. Yassine, K. Ghali, N. Ghaddar, I. Srour, G. Chehab, A numerical modeling approach to evaluate energy-efficient mechanical ventilation strategies, Energy and Buildings 55 (2012) 613-618.

[25] M. Hourani, K. Ghali, N. Ghaddar, Effective desiccant dehumidification system with two-stage evaporative cooling for hot and humid climates, Energy and Buildings 68 (2014) 329-338.

[26] W. Lin, Z. Ma, M. I. Sohel, P. Cooper, Development and evaluation of a ceiling ventilation system enhanced by solar photovoltaic thermal collectors and phase change materials, Energy Conversion and Management 88 (2014) 218-230.

[27] M. El-Refaie, S. Kaseb, Speculation in the feasibility of evaporative cooling, Building and Environment 44 (2009) 826-838.

[28] H. Zhang, E. Arens, C. Huizenga, T. Han, Thermal sensation and comfort models for non-uniform and transient environments, part II: Local comfort of individual body parts, Building and Environment 45 (2010) 389-398.

[29] M. Salloum, N. Ghaddar, K. Ghali, A new transient bioheat model of the human body and its integration to clothing models, International Journal of Thermal Sciences 46 (2007) 371-384.

[30] R. Kulkarni, S. Rajput, Comparative performance of evaporative cooling pads of alternative materials, International Journal of Advanced Sciences and Technologies, 10 (2011) 239-244.

[31] M. Beccali, F. Butera, R. A. R. S. Guanella, Simplified models for the performance evaluation of desiccant wheel dehumidification, International Journal of Energy Research 27 (2003) 17-29.

[32] M. Beccali, R. S. Adhikari, F. Butera, V. Franzitta, Update on desiccant wheel model, International Journal of Energy Research 28 (2004) 1043-1049.

[33] K. Ong, C. Chow, Performance of a solar chimney, Solar Energy 74 (2003) 1-17.

[34] M. Badawiyeh, N. Ghaddar, K. Ghali, Case Study of Trombe Wall Inducing Natural Ventilation through Cooled Basement Air to Meet Space Cooling Needs, Journal of Energy Engineering (2016).

[35] R. Kessel, Concrete and Concrete Masonry, Cement & Concrete Association of New Zealand, New Zealand, 2017.

[36] Republic of Lebanon Ministry of Public Works and Transport. Energy Analysis and Economic Feasibility, UNDP/GEF, MPWT/DGU (2005) p.63.

[37] M. Thambidurai, K. Panchabikesan, K. Mohan and V. Ramalingam, "Review on phase change material based free cooling for buildings - The way toward sustainability," *Journal of Energy strorage,* vol. 4, pp. 74-88, 2015.