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A Comprehensive Study on a Latent Heat Thermal Energy Storage System and its Feasible Applications in Greenhouses

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ABSTRACT

Energy crisis is a major challenge in the current world. Latent heat thermal energy storage (LHTES) systems are known as equipment with promising performance by which thermal energy can be recovered. In the present study a comprehensive theoretical and experimental investigation is performed on a LHTES system containing PEG1000 as phase change material (PCM). Discussed topics can be categorized in three parts. At first, a one dimensional mathematical model is introduced for a heat exchanger containing flat slabs of PCM. To consider the latent heat of phase change, effective heat capacity is used in the model. Secondly, through eight experiments designed by using factorial method, effects of inlet air velocity and temperature on the outlet stream is investigated. The results proved that having a determined temperature difference between inlet air and the PCM in both hot and cold cycles can enhance the efficiency. Finally, the feasible applications of a LHTES system for controlling the temperature swing in a greenhouse is studied numerically and the results are compared with experimental values. As a result, by using this passive coolant system diurnal internal temperature can be reduced for 10°C.

1. Introduction

Since energy is produced and transferred in the form of heat in many countries, thermal energy storage (TES) deserves to be studied in detail. One of the oldest usages of TES goes back to the time when ice was provided from frozen lakes and rivers in the winter. The collected ice was then kept in well insulated warehouses in order to satisfy the needs for food conservation and air conditioning through the year. The air conditioning of the Hungarian Parliament Building in Budapest is still done by ice harvested from Lake Balaton in the winter [1].

Storing of thermal energy takes place by using a change in the internal energy like: sensible heat, latent heat or thermo-chemical [2]. Among these methods, LHTES systems have attracted considerable attention due to their high storage capacity and also their near

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isothermal operation [1,2]. PCMs can absorb or release a high amount of energy in a small temperature difference via phase change from solid to liquid or vice versa, respectively.

So many studies have been performed for the modeling of LHTES systems and so many methods have been illustrated. One of the major techniques is related to the moving boundary problems. Unknown interface of phase change and its nonlinear movements have made the moving boundary problems quite complex [3,4]. Furthermore, this formulation can produce acceptable results mostly for PCMs with single temperature of phase change. However, most of the materials are not pure and their phase change occurs in a temperature range instead of a single point. For this type of materials, as suggested by Regin [5], enthalpy formulation can be a proper choice. This formulation does not require knowing where phase change takes place. The other method (which can also be categorized as a subset of enthalpy formulation) is effective heat capacity. In this method, the correlation between the heat capacity and temperature in the phase change transition range can be obtained by using differential scanning calorimetry (DSC) analysis. Therefore the formulation can be derived without considering the phase change [6].

30-40% of energy consumption is related to the buildings [7] and the significant portion of this amount is used on hot summer days by cooling systems having a compressor. Consequently many studies have focused on the usage of LHTES systems in cooling applications which is called free cooling. In order to condition the indoor air by changing the material phase, PCM can be embedded in a heat exchanger. During the night, PCM solidifies and the energy can be released (discharge cycle), subsequently during the hot day, via a move through the heat exchanger, air is cooled and PCM melts (charge cycle) [8,9].

The results of the simulations using the empirical model presented by Lazaro *et al*. [10] showed that the capability of the same PCM to maintain temperature levels below a certain temperature depends upon the heating power. Therefore, for any application where an almost constant temperature is required, the power demand must be taken into account. To maintain a specific temperature level when the cooling demand is high, the PCM phase change temperature should be lower. On the other hand, for very low cooling demands, the phase change temperature should be close to the objective temperature level.

Greenhouses are enveloped places in which temperature of the air must be kept in a certain limitation due to agricultural requirements. Providing this temperature limitation for crop thermal comfort is a major challenge, since the coverings need to allow light into the structure, conversely they cannot insulate very well and as a consequence unwanted heat loss/gain occurs. Lazaar *et al*. [11] conducted an experimental study to evaluate the performance of a LES unit inside a tunnel greenhouse. A shell and tube heat exchanger containing 10 kg of $CaCl_2.6H_2O$ as PCM was used in this work. They proved that by using the built LHTES system, the air temperature inside the greenhouse can be reduced by a difference of 5°C to 8°C in comparison to a greenhouse without the storage system.

Bouadila *et al*. [12] made an experimental study to evaluate the nighttime recovered heat of the solar air heater with latent heat storage collector (SAHLSC) in an east-west oriented greenhouse. It was shown that by using this passive heating system, 31% of the total heating requirements can be provided. Furthermore, they concluded that the payback period of the proposed system is approximately 5 years, if the passive heater is used only three months a year.

Chengchu Yan *et al*. [13] did a study on a seasonal cold storage in which stored cold energy in winter was used for free cooling application of a building with total gross area of 2000 m^2. In this application the cold energy from low temperature ambient air was stored in an underground storage tank (in the form of ice and chilled water) and released during summer days. The proposed system was able to provide 1/3 of total cooling demand of the studied building.

In the present research work, a one-dimensional model for a LHTES system containing flat slabs of PCM is presented in which effective heat capacity is used to calculate the latent heat during the phase change process. Afterwards a detailed study is performed on a small scale LHTES system by using numerical and experimental data. Finally, temperature changes of a greenhouse located in Tunisia, with and without using LHTES system, are calculated theoretically and the results are compared with experimental values.

2. Mathematical model

A greenhouse containing a LHTES part in its inside air circulating system is illustrated in Fig. 1. The mathematical model used for this greenhouse contains two parts, one for the LHTES system and the other one for the air temperature inside the greenhouse.

2.1. LHTES system

The mathematical model used for the LHTES system is based on the following assumptions:

1. Axial conduction in the air is neglected in the direction of the flow. This assumption is verified by the fact that the Peclet number is greater than 100 as recommended by [14] (Pe>225).
2. Temperature variations of the air normal to the flow are not considered.
3. No super cooling happens in the PCM.
4. Thermophysical properties of the PCM are constant and are the same for both phases, except the heat capacity, which is a function of temperature. This is due to the fact that temperature variations in the system are limited.
5. Thermophysical properties of air are constant. This assumption is valid because temperature variations in the process are limited.
6. Heat transfer coefficient is the same for all the slabs.
7. Heat loss to the surrounding is negligible.

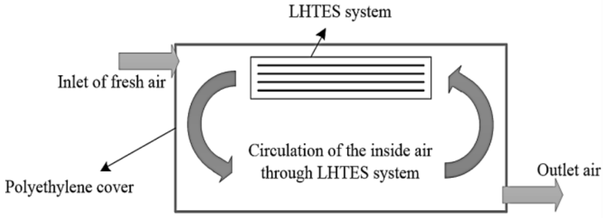

Figure 1. The schematic figure of the greenhouse.

8. Air residence time in the bed is small in comparison with the period duration.
9. Heat transfer by radiation is neglected.
10. Heat capacity and thermal resistance of PCM containers are not considered.
11. Natural convection in the melted parts of the PCM is insignificant. This assumption is valid because thickness of the flat slabs are very low(~3mm).

Based on the foregoing assumptions and a system shown in Fig. 2, the heat balance equation for the passing air through the bed of PCM is:

$$\frac{\partial T(x,t)}{\partial x}+\frac{PU_p}{vA\rho c_{pg}}T(x,t)-\frac{PU_p}{vA\rho c_{pg}}T_p(x,t)=0 \qquad (1)$$

with the boundary condition $T(0,t)=T_{in}(t)$, the solution is:

$$T(x,t)=T_p(x,t)+\left[T_{in}(x,t)-T_p(x,t)\right]\times e^{-\frac{PU_p\Delta x}{vA\rho c_{pg}}} \qquad (2)$$

By replacing $j\Delta t$ and $i\Delta x$ instead of t and x respectively, numerical form of equation (2) can be obtained as:

$$T(i+1,j)=T_p(i,j)+\left[T(i,j)-T_p(i,j)\right]\times e^{-\frac{PU_p\Delta x}{vA\rho c_{pg}}} \qquad (3)$$

In equation (3), a dimensionless parameter can be introduced:

$$\Lambda=\frac{PU_p\Delta x}{vA\rho c_{pg}} \qquad (4)$$

Λ represents the ratio of "regenerator length" to the "mass flow rate of the passing air". The transferred heat in each control volume is:

$$Q(i,j)=vA\rho c_{pg}\left[T(i+1,j)-T(i,j)\right] \qquad (5)$$

Substituting equation (3) into equation (5)

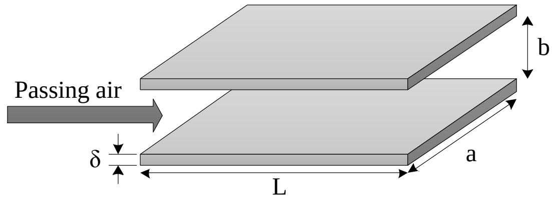

Figure 2. Configuration of the PCM slabs for establishing the energy balance equation.

gives:

$$Q(i,j)=vA\rho c_{pg}(1-e^{-\Lambda})\left[T(i,j)-T_p(i,j)\right] \qquad (6)$$

Since all the heat transferred to the air is provided by the PCM, energy balance for each control volume of the PCM can be written as:

$$q_x-q_{x+dx}-Q=m_pc_p(T_p)\frac{dT_p}{dt} \qquad (7)$$

$$k_pa\delta\frac{\partial^2T_p}{\partial x^2}dx-Q=m_pc_p(T_p)\frac{dT_p}{dt} \qquad (8)$$

$$T_p(i,j+1)=T_p(i,j)+\frac{\Delta t}{m_pc_p(T_p(i,j))}\left[vA\rho c_{pg}(1-e^{-\Lambda})\right]\left[T(i,j)-T_p(i,j)\right]+\frac{k_pa\delta\Delta t}{m_pc_p(T_p(i,j))\Delta x}\left[T_p(i+1,j)-2T_p(i,j)+T_p(i-1,j)\right] \qquad (9)$$

In the last equation a new dimensionless number can be introduced as:

$$U(T_p(i,j))=\frac{vA\rho c_{pg}\Delta t}{m_pc_p(T_p(i,j))} \qquad (10)$$

This dimensionless parameter is defined as the ratio of "mean bed temperature change" to the "mean air temperature change". The dimensionless parameters in equations (4) and (10) are the same as reduced length and utilization factor of sensible heat storage (SHS), respectively [15]. By using this similarity, another dimensionless parameter, known as reduced period, can be introduced which is a criterion of the bed heat capacity:

$$\Pi=\Lambda U=\frac{PU_p\Delta x\Delta t}{m_pc_p(T_p(i,j))} \qquad (11)$$

Finally by substituting equations (4), (10) and (11) into equation (9), equation (12) can be established as:

$$T_p(i,j+1)=T_p(i,j)+\frac{\Pi(T_p(i,j))}{\Lambda}(1-e^{-\Lambda})\times\left[T(i,j)-T_p(i,j)\right]+\frac{k_pa\delta\Delta t}{m_pc_p(T_p(i,j))\Delta x}\times\left[T_p(i+1,j)-2T_p(i,j)+T_p(i-1,j)\right] \qquad (12)$$

In the all forgoing equations U_p is defined as:

$$\frac{1}{U_p}=\frac{1}{\frac{1}{h}+\frac{\delta/2}{k_p}} \qquad (13)$$

2.2. Air temperature inside the greenhouse

To calculate the variations of air temperature inside the greenhouse, these assumptions are employed:

1. Temperature gradient inside the greenhouse is ignored.
2. Internal air is completely dry and does not have any moisture.
3. Greenhouse is empty.
4. Solar radiation is the same for all the faces of the greenhouse.

In this model the enthalpy changes of the inside air is the outcome of solar radiation, heat exchange with the inlet air, heat exchange with environment through polyethylene cover and finally heat exchange with the LHTES system [16]:

$$\begin{aligned} Vc_{pg}\rho\frac{dT_G}{dt} &= \sum_{n=1}^{5}A_n\tau_G\gamma_G S_n \\ &-A_G U_G(T_G-T_E) \\ &-m_{ACH}c_{pg}(T_G-T_E) \\ &-m_{LHTES}c_{pg}(T(0,t)-T(L,t)) \end{aligned} \qquad (14)$$

To obtain the temperature changes of the inside air, equations (3), (12) and (14) must be solved simultaneously. To this end, the outlet air temperature of the LHTES system $(T(L,t), t=(k-1)\Delta t)$ is set equal to the inside air temperature of the greenhouse (T_G^{k-1}), then by considering solar radiation, heat exchange with inlet air, heat exchange with environment through polyethylene cover, and heat exchange with the LHTES system, T_G^k is obtained. At the end by putting $T_G^k=T(0,k\Delta t)$, equation (14) changes to:

$$\begin{aligned} Vc_{pg}\rho\frac{dT_G}{dt} &= \sum_{n=1}^{5}A_n\tau_G\gamma_G S_n \\ &-A_G U_G(T_G-T_E) \\ &-m_{ACH}c_{pg}(T_G-T_E) \\ &-m_{LHTES}c_{pg}(T_G^k-T_G^{k-1}) \end{aligned} \qquad (15)$$

Using backward difference in equation (15) produces:

$$\begin{aligned} T_G^k = & \frac{Vc_{pg}\rho+m_{LHTES}c_{pg}\Delta t}{Vc_{pg}\rho+A_G U_G\Delta t+m_{ACH}c_{pg}\Delta t+m_{LHTES}c_{pg}\Delta t}\times T_G^{k-1} \\ &+\frac{A_G U_G\Delta t+m_{ACH}c_{pg}\Delta t}{Vc_{pg}\rho+A_G U_G\Delta t+m_{ACH}c_{pg}\Delta t+m_{LHTES}c_{pg}\Delta t}\times T_E^{k-1} \\ &+\frac{\Delta t\sum_{n=1}^{5}A_n\tau_G\gamma_G S_n}{Vc_{pg}\rho+A_G U_G\Delta t+m_{ACH}c_{pg}\Delta t+m_{LHTES}c_{pg}\Delta t} \end{aligned} \qquad (16)$$

3. Experimental setup

For the experimental analysis, a small-scale prototype regenerator, containing about 200 grams of PCM packed in aluminum sheets was set up. The schematic diagram of the built LHTES system is shown in Fig. 3. The pouches made of aluminum coated with polyethylene films were embedded in bed parallel to each other with 13 mm gap between them. Because of the circular cross-section of the bed, which is shown in Fig. 4, several PCM containers with different dimensions were made. The specifications and numbers of the flat slabs are reported in the Table 1. The bed container was a PVC tube 85 mm in ID, 5 mm in wall thickness and 300 mm in length. Two K-type thermocouples were affixed to measure the inlet and outlet temperatures. In this prototype PEG1000 was used as the PCM. Fig. 5 shows the result of the DSC analysis for PEG1000.

Table 1
Specifications of the flat slabs.

Dimensions of flat slabs [mm]	Mass of the injected PCM in each container [g]	Numbers of each container
3×38×210	26	2
3×53×210	38	2
3×75×210	53	1

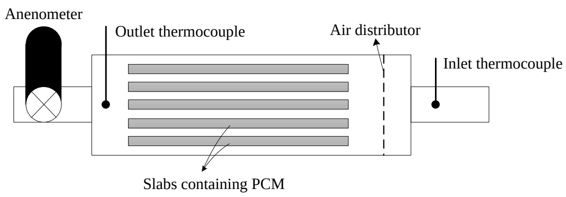

Figure 3. The schematic diagram of the built LHTES system.

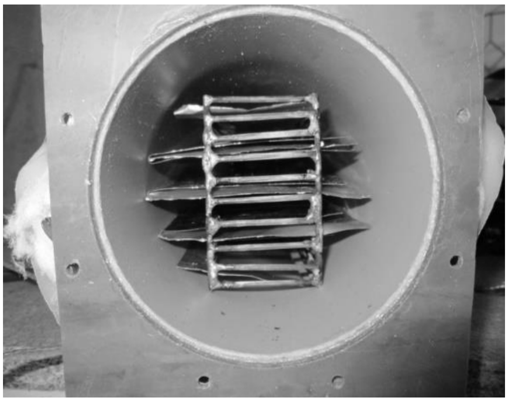

Figure 4. Circular cross section of the bed.

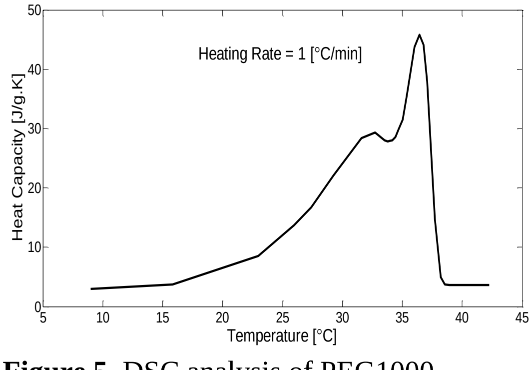

Figure 5. DSC analysis of PEG1000.

4. Results

4.1. Parametric study of the LHTES model

Simulations are carried out at different inlet temperatures and for a range of mass flow rates. Effects of the inlet temperature and mass flow rate on the charge (hot) and discharge (cold) cycles are shown in Figs. 6-9.

As shown in these figures, temperature of the inlet air plays an important role in the time needed for the process termination. In the hot cycle, higher temperatures and in the cold cycle, lower temperatures reduce the time needed for the process termination.

According to Fig. 8 and Fig. 9, higher velocities for the inlet air make the time needed for the process termination shorter.

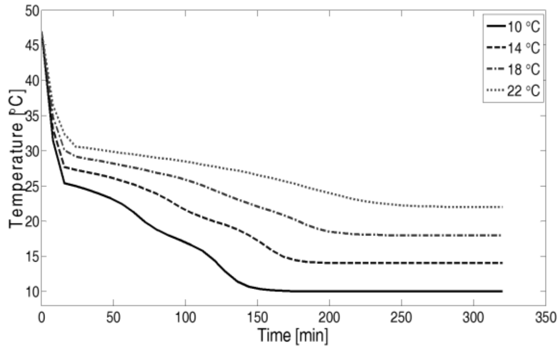

Figure 6. Temperature distribution of the cold cycle at different inlet temperatures.

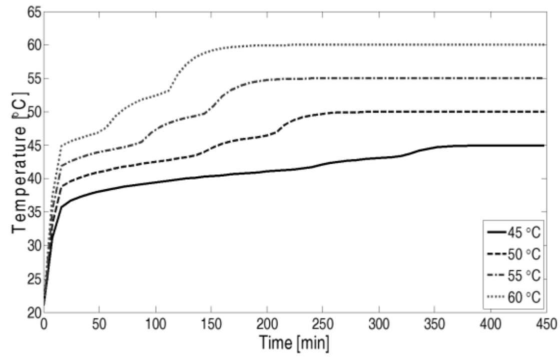

Figure 7. Temperature distribution of the hot cycle at different inlet temperatures.

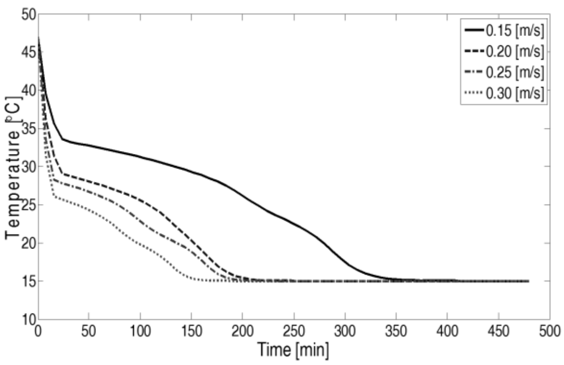

Figure 8. Temperature distribution of the cold cycle at different inlet air velocities.

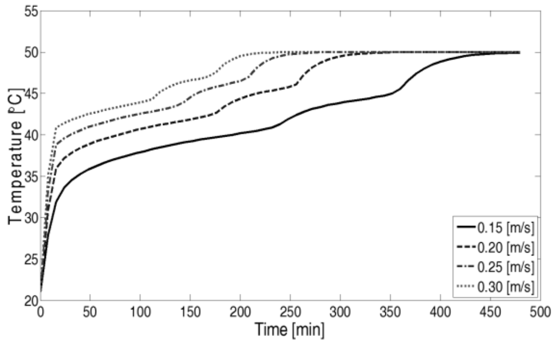

Figure 9. Temperature distribution of the hot cycle at different mass velocities.

But as shown in the forgoing plots, in higher velocities this effect becomes weaker.

In order to evaluate the performance of the built LHTES system, the following correlation was used for efficiency:

$$E = \frac{m_g c_{pg} \int_0^{\tau} (T_{hi} - T_{he}) dt}{m_g c_{pg} (T_{hi} - T_{ci}) \tau} \tag{17}$$

in which τ stands for the process duration. Effects of dimensionless parameter and process duration on the LHTES efficiency are shown in Fig. 10 and Fig. 11. As can be seen, by reducing the process duration, efficiency is enhanced. In this part, there is a tradeoff between complete phase change and efficiency, which must be considered. Because shortening the process duration makes the aborted phase change more possible and this phenomenon is not only not suitable for some applications like free cooling, but it makes the required energy used by the active coolant systems more.

As shown in Fig. 10, process duration has a significant effect on the efficiency. So for evaluating the effect of Λ on the efficiency, considering a constant criterion for the process duration seems to be necessary. Thus for all of the calculations, process termination, i.e., the time in which inlet and outlet temperatures become the same, was considered for the process duration.

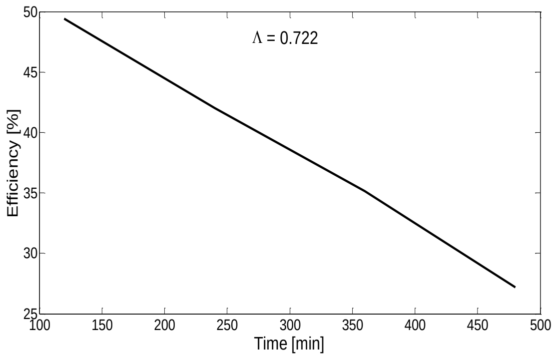

Figure 10. Effect of process duration on efficiency in constant Λ.

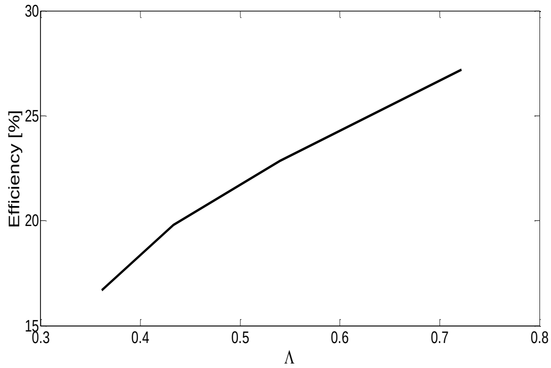

Figure 11. Effect of Λ on efficiency in the constant period (termination of the processes is considered).

As shown in Fig. 11, by increasing Λ, efficiency can be improved. It means that in the same length reducing the inlet air velocity is the solution to improve the performance of the LHTES system.

4.2. Validation of the LHTES model

In order to validate the written correlations for the LHTES, results of two experiments which were performed with the built prototype have been compared with numerical results. These comparisons are shown in Fig. 12. As can be seen there is a good consistency between the theoretical and the experimental results. The existing discrepancies are mostly related to the non ideal distribution of PEG1000 in the aluminum pouches, existence of air in the pouches, non ideal insulation of the bed and also non ideal distribution of the air throughout the circular cross section of the bed.

4.3. Experimental results

In this part eight experiments were designed with Factorial method, four experiments for the cold cycle and four experiments for the hot cycle, in which effects of operational parameters, such as the temperature of the inlet air and its velocity, on the process duration and efficiency were investigated. Tables 2 and 3 contain the related data and results for each cycle and Fig. 13 represents the experimental outputs completely. According to the illustrated results for the hot cycles in Fig. 13(a), the outlet temperature never reaches to the inlet temperature due to the heat loss and non ideal insulation.

According to the Table 2, the following results can be concluded for the cold cycle:

1. In the cold cycle, increasing the passing air velocity and reducing its temperature are two possible ways to shorten the process duration.
2. Increasing passing air velocity reduces the efficiency. It is probably related to the shortened residence time of the air in the bed.
3. In the cold cycle, increasing inlet air temperature reduces the efficiency.

According to Table 3, the following results can be concluded for the hot cycle:

1. In the hot cycle, increasing passing air velocity and increasing its temperature are two possible ways for shortening the process duration.
2. Increasing passing air velocity reduces the efficiency. It is probably related to the shortened residence time of the air in the bed.

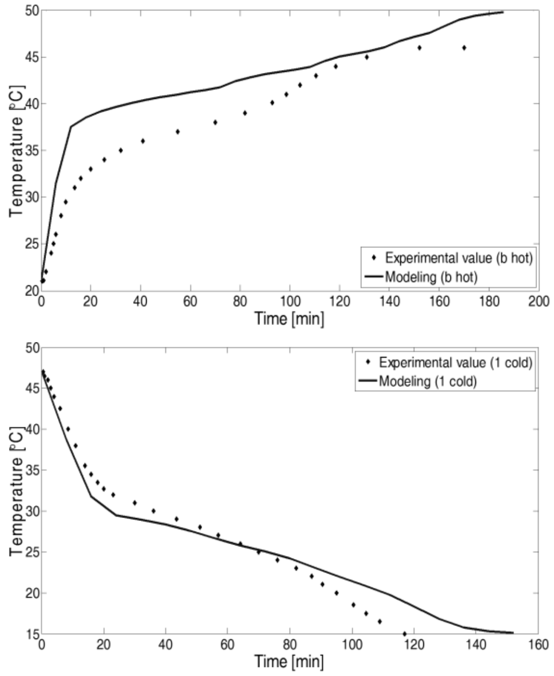

Figure 12. Comparison between theoretical and experimental results.

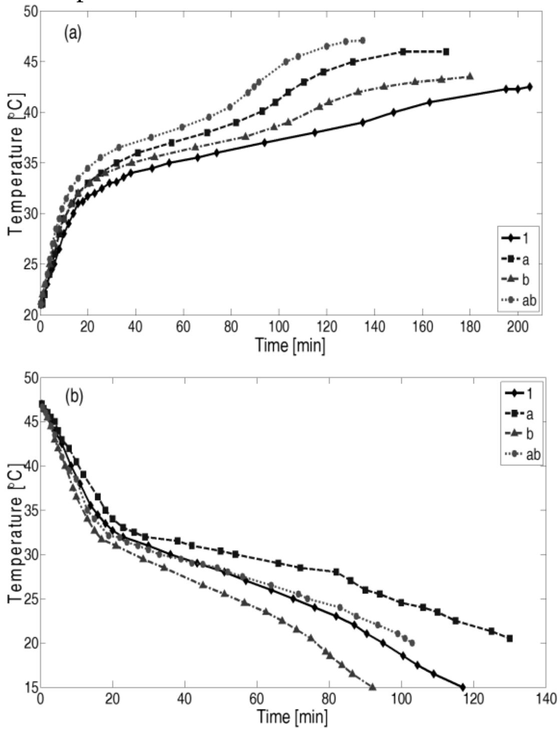

Figure 13. Experimental results: (a) hot cycles, (b) cold cycles.

3. In the hot cycle, increasing the inlet air temperature increases the efficiency.

As can be seen there is consistency between the numerical and experimental results and analyses.

Table 2
Cold cycle related data and results.

Experiment	1	a	b	ab
Inlet air temperature [°C]	15	20	15	20
Inlet air velocity [m/s]	4	4	5	5
Process termination [min]	117	130	92	103
$Q_{\max}\,[\mathrm{J}] = m_g c_{pg}(T_{initial} - T_{inlet}(t))\tau$	265973	249123	262192	250229
$Q_{real}\,[\mathrm{J}] = m_g c_{pg}\int_0^{\tau}(T_{out}(t) - T_{inlet}(t))d\theta$	97187	85728	95363	80249
Efficiency [%]	36.5	34.4	36.37	32.07

Table 3
Hot cycle related data and results.

Experiment	1	a	b	ab
Inlet air temperature [°C]	50	55	50	55
Inlet air velocity [m/s]	4	4	5	5
Process termination [min]	205	152	180	135
$Q_{\max}\,[\mathrm{J}] = m_g c_{pg}(T_{initial} - T_{inlet}(t))\tau$	410010	374700	473088	415991
$Q_{real}\,[\mathrm{J}] = m_g c_{pg}\int_0^{\tau}(T_{out}(t) - T_{inlet}(t))d\theta$	191687	187290	198437	193702
Efficiency [%]	46.7	49.98	41.94	46.56

4.4. The feasibility study on the possible applications of LHTES systems in greenhouses

4.4.1. Simulation of the greenhouse

To simulate a greenhouse using LHTES system for passive cooling, equations (3), (12) and (16) must be solved simultaneously. For the case in which LHTES system does not exist in the greenhouse, solving the equation (16) with $m_{LHTES} = 0$ is enough. As mentioned in the forgoing parts, temperature of the environment and also solar radiation, have major effects on the final answer. Fig. 14(a) illustrates the variations of the global solar radiation and ambient temperature as a function of time in 3 days in Borj Cedria area, near the city of Tunis in Tunisia. Other necessary properties are listed in Table 4. To validate the written correlation describing the variations of the internal air (in the absence of PCM storage system), model outputs are compared with the experimental values reported in [11] for greenhouse placed in Tunisia and the results are reported in Fig. 14(b). As can be seen the promising consistency guarantees the accuracy of the model.

$CaCl_2.6H_2O$ was used by Lazaar *et al.* [11] as PCM in their experimental quest for applications of passive cooler in greenhouses. Our theoretical results with

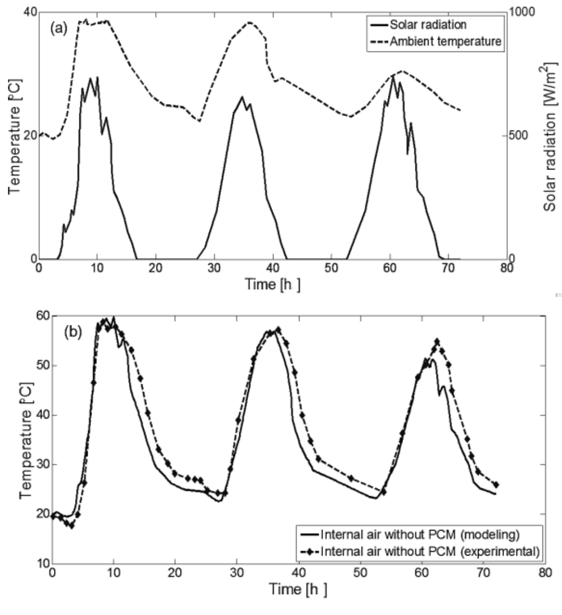

Figure 14. (a) Solar radiation and ambient temperature variations and (b) comparison between theoretical and experimental values for internal air without using PCM.

PEG1000 as PCM have been compared with values reported by Lazaar. The discrepancies are illustrated in Fig. 15. Although differences exist between these two studies, like different types of PCMs ($CaCl_2.6H_2O$ vs. PEG1000) and different geometries of the heat-exchangers (shell & tube vs. flat slabs), it can be concluded that the calculated values for air temperature inside the greenhouse are not irrational.

Temperature changes of the inside air of the greenhouse, with and without using PCM, are compared in Fig. 16. As long as $T_G < T_E$, heat exchange with environment and inlet fresh air accelerates the increasing trend of T_G. This phenomenon continues up to the time T_G=T_E. After equivalency of T_G and T_E, just solar radiation accelerates this trend and all the other factors weaken it. This increasing trend lasts as long as the heat loss to the environment and inlet fresh air equals the heat gain via radiation. When the LHTES system exists in the greenhouse, via a phase change from solid to liquid, it absorbs the excessive heat and temperature increase becomes weaker. During the night, LHTES system delivers the absorbed heat and prevents sudden temperature drop. It also becomes ready for the next day by phase change from liquid to solid.

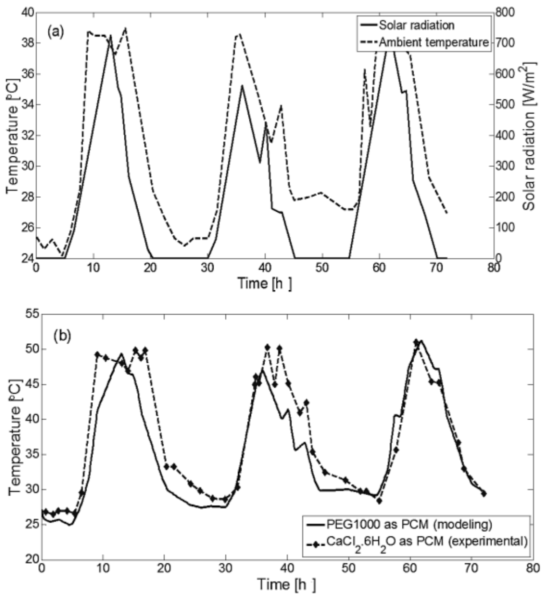

Figure 15. (a) Solar radiation and ambient temperature variations and (b) comparison between theoretical and experimental values for internal air using PCM.

4.4.2. Effect of mass flow rate on air temperature inside the greenhouse

The air temperature inside the greenhouse has been calculated for different mass flow

Table 4
Other parameters for the greenhouse and the LHTES system.

Description	Value
Greenhouse volume [m^3]	3 (height)×1.5(width)×2(length)
Cover	Single polyethylene
	$\tau = 0.6$
	$\gamma = 0.4$
	$U_G = 8$
Air	$\rho_{air} = 1.12$ [kg/m^3]
	$c_{pg} = 1005$ [J/kg.°C]
Air change per hour(ACH)	1
Number of slabs	16
Slab sizes [m]	0.42×0.205×0.003
Air vel. [m/s]	1.335
PCM	$\rho = 1093$ [kg/m^3]
	$\Delta T_m = 33-40$ [°C]
	Total mass=10 kg

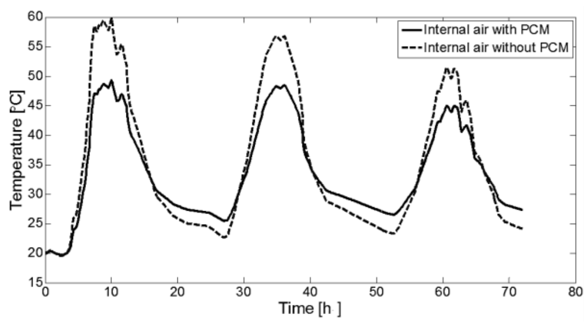

Figure 16. Comparison of internal air with and without using PCM.

rates of the circulated air (Fig. 17). As it was predictable from the forgoing parts, temperature reduction in low velocities is more tangible in charge period, since this lower mass flow rate increases the residence time of the passing air inside the heat exchanger. Another positive point is that, low velocities reduce the power consumption of the fan. It is noted that in discharge period, since cold air availability is limited, higher velocities are suggested.

5. Conclusions

With the importance of thermal energy, there is a major need to expand the optimizing utilities. In the present research work, by using experimental and numerical studies, a small scale built heat exchanger packed with flat slabs of PCM has been evaluated comprehensively. A one dimensional model considering the axial conduction in the PCM has been validated by experimental values gained from the built prototype. Experimental investigations on the effects of inlet temperature and velocity of the passing air have proved that process termination and efficiency can be affected significantly, i.e., higher temperature difference can shorten the process termination and enhance the efficiency, while higher velocities can reduce the process duration and efficiency. Existence of the "greenhouse effect" in greenhouses leads to a peak in diurnal temperature. To evaluate the applicability of LHTES systems as a solution for this problem, a scale up study has been performed numerically and it has been proved that by using PCMs, the excess unwanted heat during the hot days can be absorbed and the internal air can be cooled for 10°C during the hottest hours. For further investigations, more realistic experimental studies may be suitable for those who are interested.

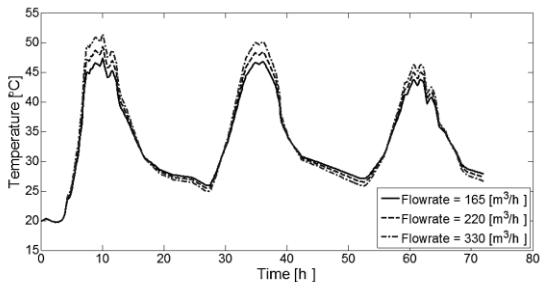

Figure 17. Effect of mass flow rate on internal temperature.

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Nomenclature

a	Width of flat slabs [m]
A	Area of cross section of duct= ab [m^2]
A_i	Area of each face of the greenhouse [m^2]
A_G	Total area of the greenhouse faces [m^2]
b	Air gap between parallel slabs [m]
c_p	Heat capacity of PCM [J/kg°C]
c_{pg}	Heat capacity of air [J/kg°C]
Δx	Spatial length [m]
Δt	Time step [sec]
E	Efficiency
h	Heat transfer coefficient [W/m^2°C]
i	Spatial step counter
j	Time step counter
k_p	Thermal conduction of PCM [W/m^2°C]
L	Length of the Bed [m]
m_{ACH}	Mass flow rate of air change [kg/s]
m_{LHTES}	Mass flow rate of air in LHTES [kg/s]
m_p	Mass of PCM [kg]
P	Perimeter [m]
Pe	Peclet number
Pr	Prantl number
Q	Transferred heat [J]
S_i	Solar radiation [w/m^2]
t	Time [sec]
T	Temperature of the air in LHTES [°C]
T_E	Environmental temperature [°C]
T_G	Air temperature of the greenhouse [°C]

T_i	Initial temperature [°C]
T_p	PCM temperature [°C]
U	Dimensionless parameter of regenerator
U_G	Natural heat transfer coefficient [W/m^2°C]
U_p	Overall heat transfer coefficient [W/m^2°C]
V	Total volume of the greenhouse [m^3]
x	Length variable [m]
τ	Process termination [s]
τ_G	Transmittance of the greenhouse cover to the direct solar radiation
ρ	Density of air [kg/m^3]
β	Thermal Expansion [K^{-1}]
α	Thermal diffusivity [m^2/s]
v	Air velocity [m/s]
δ	Thickness of slabs [m]
Λ	Dimensionless parameter of regenerator
Π	Dimensionless parameter of regenerator
γ_G	Constant of the proportion of solar radiation entering the greenhouse

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