Numerical Study OF Thermal Peroformances of a PCM-Air Solar Heat Exchanger

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Abstract: Thermal energy storage got a significant role in the solar energy conservation in order to expand its use over time. To exploit solar energy continuously, we require a storage energy system. Phase Change Material (PCM) is used in this kind of systems in order to store a great amount of thermal energy. In fact, Latent heat storage in a PCM is very interesting because of its high-energy storage density and its isothermal behavior during the phase change process. Hence, heat accumulated during sunshine period, can be restituted to be used for air conditioning purposes in buildings. In this perspective, we propose in this work a numerical study based on an enthalpy formulation to investigate the solidification of a PCM in a heat exchanger with and without fins. This numerical approach gives simultaneously the temperature distributions in the PCM storage system and temporal propagation of the solidification front during the solidification of the PCM when it is exposed to a cold air flow. Also, we give in this study the transient evolution of the longitudinal air temperature profiles.

Key-Words: PCM, Numerical study, thermal energy, latent heat, fin

Nomenclature

dt	time step
Tair	temperature of air at node, K
Cp_{air}	heat capacity of air, J. $K^{-1}.kg^{-1}$
h	heat transfer coefficient $W.m^{-2}.K^{-1}$
w	air velocity, $m.s^{-1}$
T_0	inlet air temperature
ex	width of the computational domain
ey	thickness of the computational domain
ez	length of the computational domain
X,Y,Z	dimensionless coordinates system
i	width index
j	thickness index
k	lengh index
Cp(T)	heat capacity per unit volume including the phase change, $J.m^{-3}.K$
Cp_s	solid phase heat capacity per unit volume, $J.m^{-3}.K$
Cp_l	liquid phase heat capacity per unit volume, $J.m^{-3}.K$
L	latent heat, $J.kg^{-1}$
T_{liq}	temperature of liquid interface, °C
T_{sd}	temperature of solid interface, °C
f_s	solid fraction
T_{air}	initial temperature of air, °C
T_{PCM}	initial temperature of PCM, °C

Greek symbols

ρ_{air}	air density, $kg.m^{-3}$
λ_{air}	heat conductivity of air, $W.m^{-1}.K^{-1}$
$\rho(T)$	density including the phase change, $kg.m^{-3}$
ρ_s	solid phase density, $kg.m^{-3}$
ρ_l	liquid phase density, $kg.m^{-3}$

1 Introduction

Developing efficient and inexpensive energy storage devices is very important to develop new sources of energy. The Thermal Energy Storage (TES) can be defined as the temporary storage of thermal energy at high or low temperatures. The TES is not a new concept, and it has been used for centuries. Energy storage can reduce the time or rate mismatch between energy supply and energy demand, and it plays an important role in energy conservation. There are three modes of (TES): sensible heat storage (SHS), latent heat storage (LHS), and bond energy storage (BES) **[1]**. Due to rising energy costs, thermal storages systems designed for the heating and cooling of buildings are becoming increasingly important **[2-4]**. One of the most preferable storage techniques is Latent Heat Thermal Energy Storage (LHTES) by Phase Change Materials (PCMs) due to its important energy storage density and its high isothermal storage process **[5-6]**.

Every LHTES should include three main components: an appropriate PCM in the required temperature range, a container for the stocking substance, and an appropriate carrying fluid for an effective heat transfer from the heat source to the heat storage. Moreover, the PCMs require an important heat exchange area because of its low thermal conductivity. One method is to increase the heat transfer surface area by employing finned surfaces **[7-10]**. Many numerical and analytical models of PCM solidification in finned PCM-Air exchangers have been published in order to evaluate their performance. Mosaffa et al. **[7]** studied analytically the PCM solidification process in a shell and tube finned thermal energy storage for air conditioning systems. This analytical solution is compared to that obtained via a two-dimensional numerical method based on an enthalpy formulation for prediction of the solid–liquid interface location. Furthermore, the variation PCM solid fraction with time is compared for the cylindrical shell and rectangular container storage arrangements, with the same volume and heat transfer surface area. The results show that the solidification of the PCM in cylindrical shell storage is more rapidly than in rectangular container storage. In addition, the solid fraction of the PCM increases more quickly when the cell aspect ratio is small. The effects of air velocity and inlet air temperature on the performance of the thermal storage are analyzed. It has been found that the effect of inlet air temperature is more significant than that of air velocity on the outlet temperature. Al Abidi et al. **[8]** studied numerically the PCM solidification in a triplex heat tube exchanger with internal and external fins. Different design parameters as heat transfer enhancement techniques, which included the fin length, number of fins, fin thickness, and PCM unit geometry, were analyzed. Simulated results show that such parameters had significant influence on the time to cool the PCM completely. The effect of fin thickness was smaller than that of the fin length. The number of fins and the number of PCM unit geometries have a similar effect of the solidification time. The results indicate that, for a thickness of fin equal to 1 mm, a complete solidification can be achieved by 8 PCM unit geometries in a less time (35%) when compared to the other cases. Bauer **[9]** developed approximate analytical solutions for the solidification of PCMs in fin geometries using effective thermo-physical properties. He studied two fundamental geometries which are the finned plane wall and a single tube which is radial-finned on the outside. Rostamizadeh et al. **[11]** developed a numerical model of energy storage in a rectangular container of PCM **[12-14]**, based on an enthalpy formulation and the effect of PCM thickness on temperature distribution and melting fraction was investigated. Thus, these researchers established that 5 mm is the best thickness of the PCMs. Besides the study results show that the PCM mass and the melting time verify a linear relationship. Ismail et al. **[5]** analyzed a numerical and experimental study on the solidification of PCM around a vertical axially finned isothermal cylinder. The model is based upon the pure conduction mechanism of heat transfer. From the given results, it can be noticed that the fin thickness hasn't any important influence on the time of solidification, indeed the time for full solidification and the solidification rate is strongly affected by the fin length and the number of fins. The difference of temperature has a reverse impact on the solidification of PCM, and the full solidification time tends to decrease when the temperature difference increases.

In this article, we present a numerical study for the three-dimensional solidification process of a PCM (c_{18}) in a PCM-Air heat exchanger. The choice of the form of the storage unit is dictated by the need to improve the exchange surface during the heat restitution. We deliberately chose a gear geometry side of convective heat transfer. A comparative study is presented for a rectangular container with straight toothed in the heat exchanger with and without fins (Fig.1) in order to show the fins' influence on the solidification. This numerical approach gives simultaneously the temperature distributions in the PCM storage system, temporal propagation of the solidification front during the solidification of the PCM when it is exposed to a cold air flow, and the transient evolution of the air temperature profile.

2. Methods

In this work, the computational domain relates to the energy storage unit. It comes to a metal container filled with PCM and whose wall thickness is assumed to be negligible (**Fig.1**).

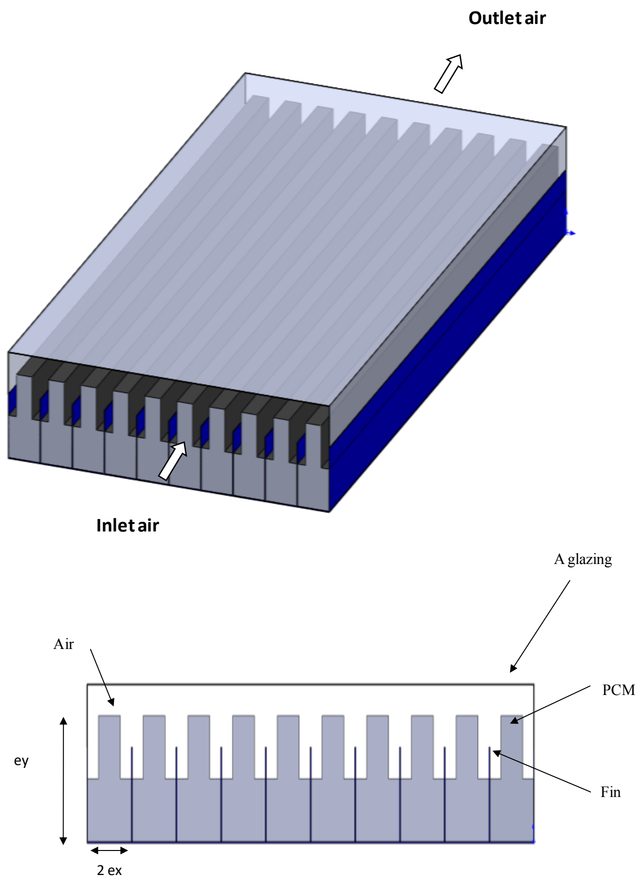

Fig. 1. Studied configuration.

The computational domain's dimensions are ex=0.025m, ey=0.2m and ez=2m Taking into account the symmetry conditions in the storage units, just undertake the calculations in a volume forming a half tooth (**Fig.2**).

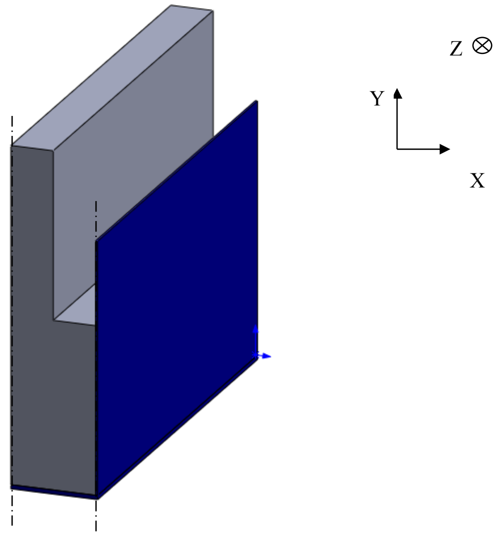

Fig.2. Computational domain.

The mathematical model was established under the following assumptions:

(1) The effects of natural convection within the solidification are negligible and can be ignored.
(2) The liquid PCM and the fins are initially at 45°C.
(3) The calculation domain includes the container and the fins.
(4) The PCM is homogenous and isotropic.
(5) Thermo-physical properties of the PCM are different for solid and liquid phases but are independent of the temperature.
(6) The heat transfer through air is considered unidirectional along (OZ).
(7) Thermal conductivity of the air in the direction of the HTF flow is ignored.

2.1 Heat exchanger model as airflow in a duct in (oz)-direction

The heat exchanger is modeled as duct with airflow where the PCM have a temperature $\overline{T}_{PCM}$

$$\varphi_{air} Cp_{air}\left(\frac{\partial T_{air}}{\partial t} + w\frac{\partial T_{air}}{\partial z}\right) = \lambda_{air}\frac{\partial^2 T_{air}}{\partial z^2} + \frac{h.dA\left(\overline{T}_{PCM} - T_{air}\right)}{dV} \quad (1)$$

$\lambda_{air}\dfrac{\partial^2 Tair}{\partial z^2}$ is considered negligible.

$\overline{T}_{PCM}$: Average temperature in section crossing node 'k'

The numerical solution of the governing differential equation is given by finite difference method using the upwind spatial discretization schema.

$$\left(\frac{1}{\Delta t}+\frac{w}{\Delta Z}+\frac{h}{\varphi_{air}Cp_{air}}\frac{\Delta A}{\Delta V}\right).T_{air}^{\,t+\Delta t}(k)= \qquad (2)$$

$$\frac{w}{\Delta Z}T_{air}^{\,t}(k-1)+\frac{1}{\Delta t}T^{t}_{air}(k)+\frac{h}{\varphi_{air}Cp_{air}}\frac{\Delta A}{\Delta V}\bar{T}^{t}_{PCM}(k)$$

Where;

$T_{air}(1)=T_0$

ΔA : The exchange finite area.

ΔV : The finite volume of airflow in duct.

2.2 PCM model with finite volume discretization method

The Equation of heat transfer through the PCM is written in a three-dimensional Cartesian coordinates system. Considering that the heat transfer process is controlled only by pure conduction, one can write the conduction equation as follows:

$$\rho(T)Cp(T)\frac{\partial T}{\partial t}=\lambda(T)\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \qquad (3)$$

Where;

$$\rho(T)Cp(T)=\begin{cases}\rho_s Cp_s & T\le T_{sd}\\ \rho_l Cp_l & T\ge T_{liq}\\ \dfrac{\rho_s L}{T_{liq}-T_{sd}} & T_{sd}\le T\le T_{liq}\end{cases} \qquad (4)$$

$$\lambda(T)=\begin{cases}\lambda_s & T\le T_{sd}\\ \lambda_l & T\ge T_{liq}\\ \dfrac{\lambda_l-\lambda_s}{T_{liq}-T_{sd}}(T-T_{liq}) & T_{sd}\le T\le T_{liq}\end{cases}$$

The discretization of **Eq. (3)** is realized by using the control volume technique. In this case the domain of the problem is divided into a convenient number of control volumes (40×150×200) and **Eq. (3)** is integrated over each control volume, to calculate the value of T at the time t+Δt on all points defining the computational domain. To do this we use the Douglas-Gunn modified method. It is carried in three stages and is based on the factorization of the spatial operator in three-dimensional operators. Starting from an initial solution at time t, the time level t+Δt will be reached in three stages providing three intermediate solutions T *,T **, T ***, the latter gives an approximation of $T^{t+\Delta t}$.

2.3 The initial and boundary conditions

Initially, it is assumed that the storage units are temperature T_{PCM} = 45°C, and the inlet air temperature T_{air} = 10°C. The boundary conditions of the computational domain are:

- Convection boundary conditions, for all surfaces which are in direct contact with air.
- where there are a symmetry planes: (**Fig.3**).

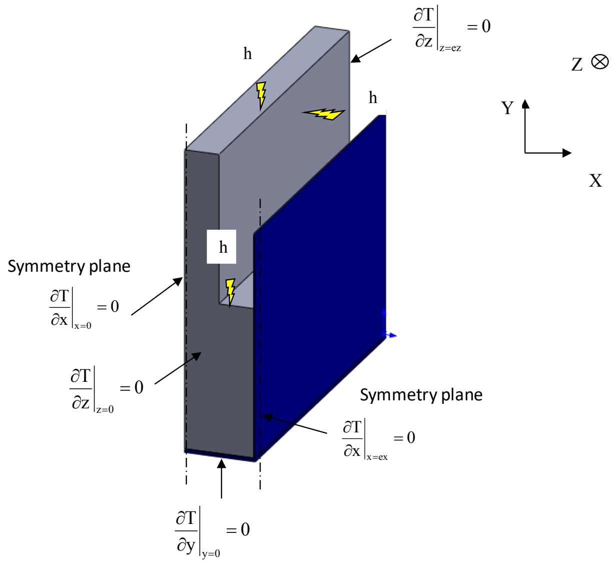

Fig. 3. Boundary conditions.

3 Verification

The performance of the presented method is verified with experimental data performed by Zivkovic and Fujii **[14]** by simulating under the same operating conditions the melting process of ($CaCl_2$ $6H_2O$) used as PCM, with thermo-physical properties as listed in **table 1**.

Table1. Thermo-physical properties of $CaC1_2.6H_2O$

Melting point [°C]		29.9
Latent Heat [kJ.kg^{-1}]		187
Density [kg.m^{-3}]:	Solid	1710
	liquid	1530
Specific heat [kJ.kg^{-1}.K^{-1}]:	Solid	1.4
	liquid	2.2
Thermal conductivity [W.m^{-1}.K^{-1}]:	Solid	1.09
	liquid	0.53

A rectangular container, made of stainless steel, with dimensions l=b=100 mm and δ=20 mm (**Fig. 4**) was filled with the calcium chloride hexahydrate and well insulated on the lateral sides. The computational model was set up to reproduce experimental conditions within the constant temperature bath. In **Figure 5**, the variation with time of numerical and experimental values of the temperature at the center of the test container is shown. It can be concluded that agreement between numerical and experimental data is well within experimental uncertainties.

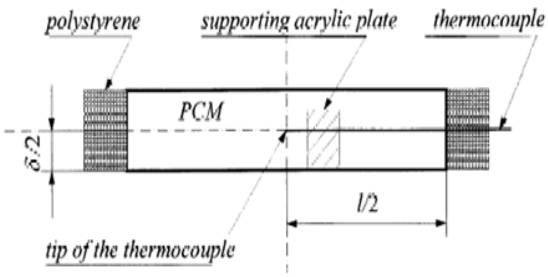

Fig. 4 Test container (Zivkovic and Fujii [**14**]).

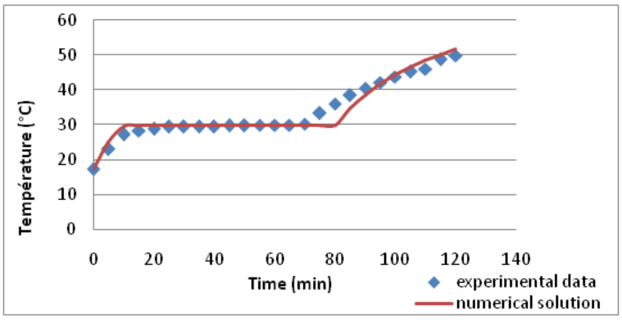

Fig. 5 Variation with time of PCM temperature at the center of the rectangular container.

4 Results and discussion

In this work, we studied two cases; the first case treats the solidification of PCM in a PCM-air heat exchanger without fins, while the 2nd case focuses on the solidification of PCM in a heat exchanger of the same type and the same geometry and under the same conditions but adding the fins as shown in **Figure 1**.

4.1 Solidification of PCM in a PCM-air heat exchanger without fins

Figure 6 shows the solidification evolution of the PCM cartographies in the middle X-Y plane of the PCM module, at different times (1/4h, 1h, 2h, and 3h). Temperature evolution goes through three distinct stages. During the first stage the PCM temperature decreases with time, until the PCM begin to solidify. Heat was extracted from PCM liquid phase by conduction from the start of cooling process to the beginning of the phase change. The material restores the energy primarily by sensible heat. During the second stage the energy is mainly changed by latent heat inside the PCM, giving progressively a solidification front. During the third stage, heat transfer was predominated by conduction from the finish of cooling process to the finish of the phase change. For all these instants, the solidification front moves from the surface of PCM, which is in contact with cold air, to the deeper zones. Comparing between these cartographies, we can notice that the amount of PCM solidified is very small. We can explain this by the fact that the thickness of the solid PCM layer becomes more important with time and so the transfer resistance increases, which inhibits the heat transfer.

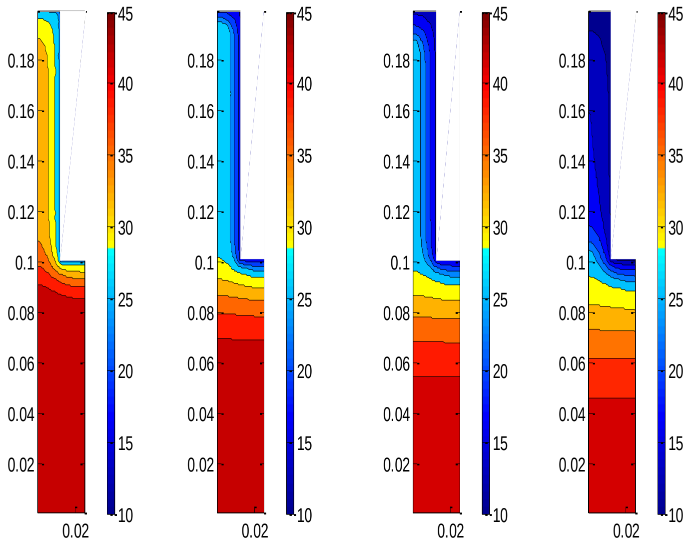

Fig. 6 Field at different times of solidification in the middle X-Y planes.

Figure 7 shows the solidification evolution of PCM in the symmetry Y-Z plane at different times (1/4h, 1h, 2h, and 3h). The part of plane situated in the depth of the PCM container remained at high temperature indicating that PCM is still at liquid state. However, the upper part of the plane is cooled more rapidly. This is more pronounced in the entrance of the duct since the inlet temperature of air is 10°C.

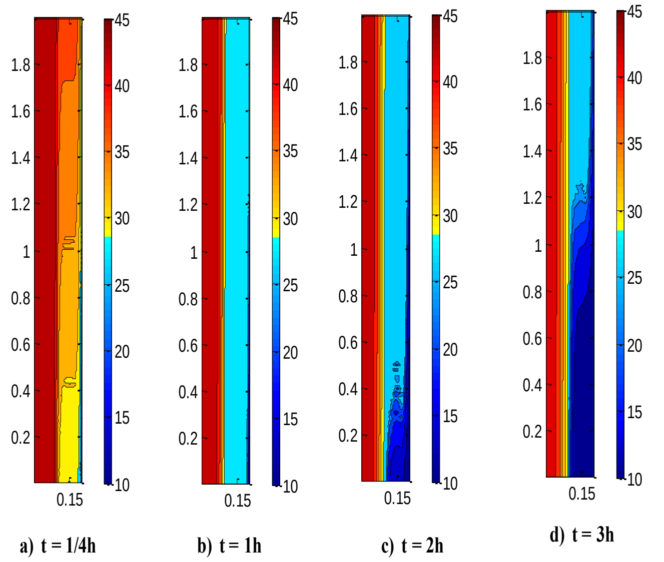

Fig. 7 Temperature fields at different time of solidification in the symmetry Y-Z plane.

Figure 8 shows the temperature fields of PCM in the planes Z=0.1, Z=0.5, Z=1 and Z=1.6m at time t = 1/4h. Comparing temperature fields in these 4 planes, we can note that the solidification of the PCM begin in the entrance side of the air. Hence in the plane (Z=0.1), the PCM is the coldest.

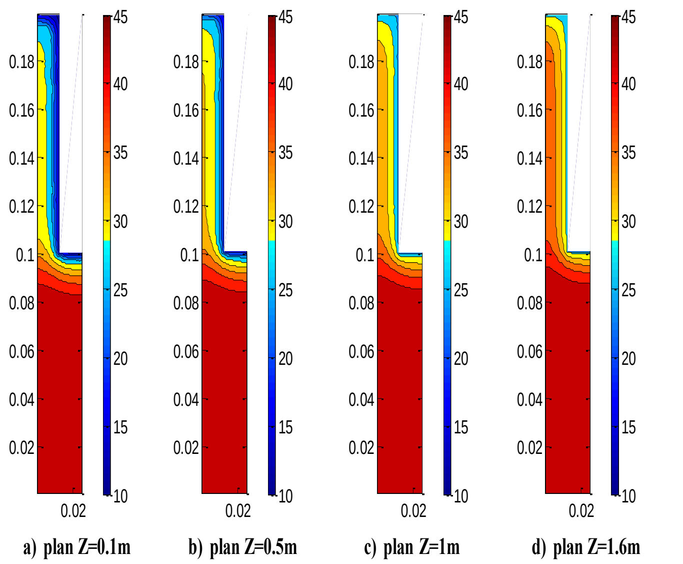

Fig. 8 Temperature fields at different longitudinal planes at time t = 1/4h.

Figure 9 shows the transient evolution of the longitudinal air temperature profiles at different times (1/4h, 1h, 2h, and 3h). It appears that the outlet air temperature decreases very rapidly with time, reaching values that have no interest in practice. In fact, the air temperature at the outlet of the exchanger does not exceed 15 ° C. To overcome this disadvantage, we propose to integrate steal flat surface fins in the PCM module to enhance heat transfer to air (**Fig.1**).

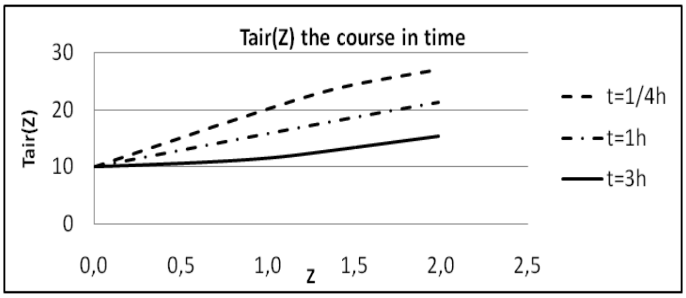

Fig. 9 Air profiles the course in time (heat exchanger without fins).

4.2. Fin effect of PCM solidification

Figure 10 compares the temporal evolution with time of the temperature field in the middle X-Y plane of the PCM module for the heat exchangers with and without fins. We can show that the solidification front moves faster in the case of the finned heat exchanger. This is caused by the fact that the fin, penetrating the PCM, well conducts the heat from the deeper region of the PCM.

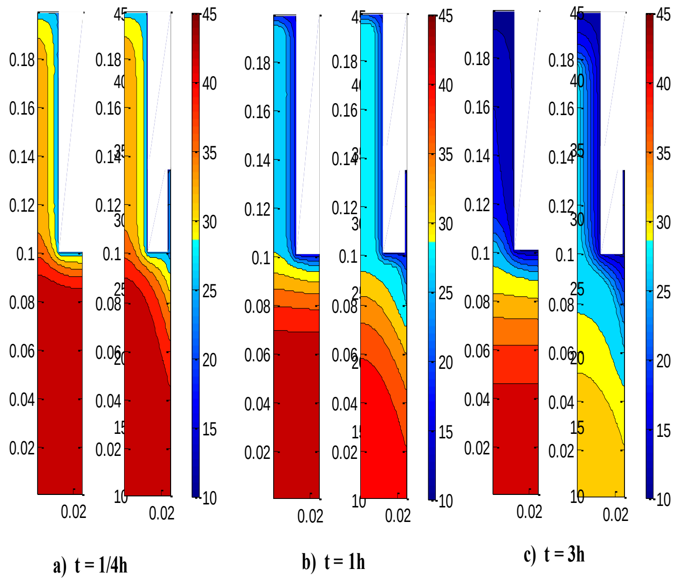

Fig. 10 Temperature fields at different times of PCM in the middle X-Y plane for the heat exchangers with and without fins.

Figure 11 shows a comparison between longitudinal air temperature profiles with and without fins. We can note that the air temperature at the outlet of the exchanger with fins is more interesting than the exchanger without fins. This shows that the presence of fins enhances the performance of the heat exchanger.

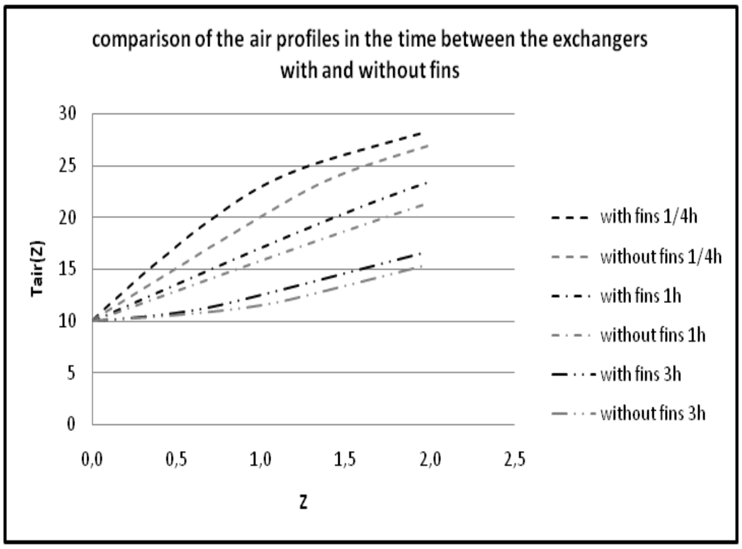

Fig. 11 Air profiles over time of exchangers with and without fins.

5 Conclusion

The purpose of this research was to provide a numerical model for the PCM-Air heat exchanger system. During the day, hot air passing through the exchanger, causes melting of the PCM, and during the night, we restitute accumulated energy by heating air injected in local. The model was based on enthalpy formation which was presented by numerical analysis. In this work we showed that heat storage in a PCM is interesting but the most interesting is the ability to extract this latent heat. In this work, we show that the presence of fins has enhanced the performance of the heat exchanger.

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