

PARALLEL PLATE CHANNELS FOR LATENT HEAT THERMAL ENERGY STORAGE

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ABSTRACT

Lately TES systems for thermal applications, such as space and water heating, air-conditioning, cooling, etc. have received much attention. In TES system, energy is stored by changing the temperature of a storage medium or employing the latent heat of Phase Change Material (PCM). For a TES system, it is important to have a charging velocity rather high, because for an assigned system size, it is possible to obtain greater amount of stored energy in less time. Therefore, the heat transfer between the working fluid and the TES system should be improved by increasing the surface exchange area.

In the present work a computational investigation on thermal energy storage system using Phase Change Material (PCM) is accomplished. The system is a set of different parallel plates, half of them are filled with PCM and in the others the working fluid, air, flows. Various configurations for different channels per unit of length (CPL) are investigated. A comparison between a direct model and a porous medium model is carried out. The porous medium is modeled with the Darcy-Forchheimer law. Numerical simulations are carried out using the Ansys-Fluent code for the direct model with 2n channels for direct and equivalent porous medium.

INTRODUCTION

Systems for the energy saving and storage are mainly used in many industrial and commercial applications to supply thermal energy when it is required. The energy demand in these applications is not always constant and therefore a Thermal Energy Storage (TES) system represents a good solution to cover such problem, in particular by means of Latent Heat Thermal Energy Storage (LHTES) systems with PCMs [1]. In LHTES systems, the energy is stored employing the latent heat of Phase Change Material (PCM) [2]. The Phase Change materials have the possibility to store the energy during the phase change process at quasi-constant temperature. The melting temperature is different in base of the PCM nature. Some review articles [3-4] examines various PCM, their thermophysical properties and the enhancement systems in thermal storage applications.

NOMENCLATURE

c, c_p	[J/kgK]	Specific heat
H	[m]	Height of a single elementary channel
H_L	[J/kg]	Latent heat of fusion
k	[W/mK]	Thermal conductivity
K	[m ²]	Porous Permeability
L	[m]	Length of a parallel plate
n	[-]	Air Channel per unit of length (CPL)
p	[Pa]	Static pressure
Pe	[-]	Peclet number of the channel
s	[m]	Thickness of a single elementary channel
t	[s]	time
T	[K]	Temperature
u	[m/s]	Air x-Velocity
U	[m]	Unit of length
v	[m/s]	Air y-Velocity
V	[m ³]	Volume
W	[m]	Width of the system
x	[m]	Cartesian axis direction
y	[m]	Cartesian axis direction
z	[m]	Cartesian axis direction
Special characters		
α_{sf}	[m ²]	Area Surface density
β	[-]	Liquid fraction
ΔT	[K]	Melting range temperature
ρ	[kg/m ³]	density
Δp	[Pa]	Air Pressure drop
ε	[-]	Air Porosity
μ_f	[m ² /s]	Dynamic viscosity of the air
Subscripts		
air		air
avg		average
$cord$		cordierite
d		dispersion
D		Darcy
eff		effective
f		Porous fluid phase, air
m		PCM melting
PCM		Phase change material
$total$		packaging
n		Air Channel per unit of length (CPL)
s		Solid phase of the porous zone

The LHTES systems are useful to store the solar thermal energy, because it is periodic with time and therefore the LHTES system stores exceed energy and release it when it is demanded. In literature there are many works about the PCM for thermal storage, in particular Kurnia et al. [5] evaluate numerically various configurations of PCM in thermal energy storage system with different configurations and for charging and discharging time. They found that the various arrangement of PCMs significantly affects the heat transfer performance. Andreozzi et al. [6] studied a TES honeycomb system with a solid matrix for solar applications at different porosities. The system is modelled as a porous medium using the local thermal non-equilibrium assumption. Moreover, a transient analysis of an high temperature thermal storage honeycomb system with parallel squared channels is numerically studied in [7], where the TES system is compared with an anisotropic porous medium model. The results showed that for high number of channels, the honeycomb system can be considered as a porous model. The combination of thermal energy storage system with phase change materials and the air as heat transfer fluid is studied in literature [8], nevertheless there are few works where the PCM and the air are arranged in a matrix system in order to increase the heat exchange area. Stathopoulos et al. [9] experimentally and numerically studied a PCM-air heat exchanger. The exchanger is composed of a set of 16 parallel placed aluminium plates, filled with paraffin. The unit was coupled with an experimental platform allowing air circulation through the plates. The results of this work showing a good agreement between the numerical and experimental apparatus.

In the present work a computational investigation on thermal energy storage system using Phase Change Material (PCM) is accomplished. The system is a set of different parallel plates, half of them are filled with PCM and in the others the working fluid, air, flows. Various configurations for different channels per unit of length (CPL) and inlet velocities are investigated. A comparison between a direct model and a porous medium model is carried out. The porous medium is modelled with the Darcy-Forchheimer law and to evaluate the heat transfer among solid, PCM and air and the local thermal non-equilibrium assumption is employed. Permeability, inertial resistant coefficient, effective thermal conductivity and interfacial heat transfer coefficient of the equivalent porous medium are evaluated by means of the direct model. Numerical simulations are carried out using the Ansys-Fluent code for the direct model with $2n$ channels for direct and equivalent porous medium. Results in terms of melting time, temperature fields, stored energy as function of time are presented and a comparison between the two models is accomplished.

GOVERNING EQUATIONS

The system under consideration is a set of different parallel plates, half of them are filled with PCM and in the others the working fluid, air, flows. The sketch of the system is depicted in Figure 1. The height of a single elementary channel is H and the thickness is $2s$ and therefore the cross section area where the fluid flows is $H \times W$. The length L of the parallel plate is 1 m.

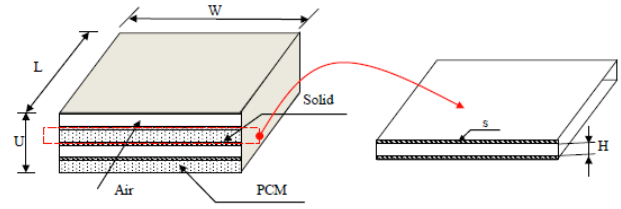


Figure 1 Sketch of the parallel plates with the single channel size.

The heat transfer fluid is air and the solid is cordierite, a ceramic material. The PCM used in this simulation is Potassium carbonate (K_2CO_3). The thermal properties of the heat transfer fluid are temperature dependent, while the solid and the phase change material are considered temperature independent. The properties are listed in Table 1. The variation of thermal properties for the air are described by the following equations [10]:

$$c_p = 1.06 \cdot 10^3 - 0.449T + 1.14 \cdot 10^{-3}T^2 - 8 \cdot 10^{-7}T^3 + 1.93 \cdot 10^{-10}T^4 \quad (1)$$

$$k = -3.93 \cdot 10^{-3} + 1.02 \cdot 10^{-4}T - 4.86 \cdot 10^{-8}T^2 + 1.52 \cdot 10^{-11}T^3 \quad (2)$$

The air through the parallel plates system while the PCM is confined inside it, therefore half of channels are closed where the PCM is blocked and the others are free to be passed through, as it showed in figure 1.

Table 1 Thermal properties of the materials

Thermal properties	Cordierite	Air [10]	K_2CO_3 [4]
Density [kg/m ³]	2300	Ideal gas law	2290
Specific Heat [J/kgK]	900	Eq. 1	1513.69[11]
Thermal Conductivity [W/mK]	2.5	Eq. 2	2
Dynamic Viscosity [kg/ms]	-	Sutherland law	-
Thermal expansion coefficient [1/K]	-	-	0.00011
Melting Heat [J/kg]	-	-	235800
Melting Temperature [K]	-	-	1170.15

Various parallel plates system for different air Channel per unit of length (CPL) are studied with the same porosity and volume. In particular, the parallel plates system with CPL equal to 1 and 4 are studied and by the results the permeability, effective thermal conductivity, local heat transfer coefficient and specific area density are calculated. The parallel plates system with different CPL is reported in figure 2.

The relation between the channel height " H " and the thickness " s " for different CPL is:

$$H_n = \frac{H}{2^{n-1}}; \quad s_n = \frac{s}{2^{n-1}} \quad \text{for } n = 1, 2, \dots \quad (3)$$

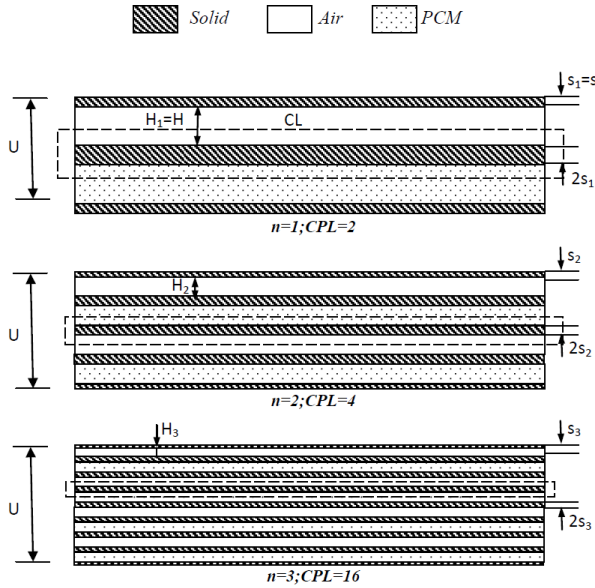


Figure 2 Sketch of parallel plates system with 2, 4 and 8 channel per unit of length (CPL).

where H and s are the values for 1 and $n = \log_2 \text{CPL}$. Therefore, the porosity for this configuration is:

$$\varepsilon = \frac{V_{\text{air}}}{V_{\text{total}}} = \frac{H_n}{2H_n + 4s_n} = \frac{H}{2H + 4s} \quad (4)$$

where V_{air} is the air volume and V_{total} is the packaging volume. The fraction of the PCM phase is equal ε because half of channels are filled with PCM and the fraction of the cordierite is equal $1-2\varepsilon$.

The relation between unit of length U , channel height and thickness is:

$$U = 2H + 4s \rightarrow \begin{cases} H = U\varepsilon \\ s = (1-2\varepsilon)/4 \end{cases} \quad (5)$$

The porosity value is set to 0.4 and the unit of length U is equal to 0.2 m.

The direct simulation of the parallel plates systems are then compared with a porous model with the same characteristics such as permeability and effective thermal conductivity. To simulate the melting and solidification of PCM the enthalpy-based method is employed [12]. In this method, during the phase change a mixed solid-liquid phase zone is present, called mushy zone. This region is described using a parameter called liquid fraction, β . Its value varies from 0 to 1 in the mushy zone, while it is zero when the zone is fully solid and it is 1 when the zone is fully liquid:

$$\begin{cases} \beta = 0 & \text{for } T < T_m - \Delta T / 2 \\ \beta = \frac{T - T_m + \Delta T / 2}{\Delta T} & \text{for } T_m - \Delta T / 2 < T < T_m + \Delta T / 2 \\ \beta = 1 & \text{for } T > T_m + \Delta T / 2 \end{cases} \quad (6)$$

where T is the local temperature, T_m is the melting temperature of the PCM, ΔT is the temperature range where the phase

change occurs. The mushy zone exists in the range temperature ΔT . In this study the ΔT is set to 2K in order to avoid numerical instability. The PCM is enclosed inside the channels and it is considered fixed without any movement, while the air passes through with a mass flux of $0.05 \text{ kg m}^{-2} \text{ s}^{-1}$.

When the number of CPL increases, it is more complex to simulate the parallel plates system, therefore an equivalent porous model is used and it is compared with the direct simulation. In fact, the only sealed up parallel plates filled up with PCM (in the others plates the air flows) can be considered as a porous medium where it is necessary to define the characteristics as effective thermal conductivity, permeability. The porous medium is anisotropic with an anisotropic effective thermal conductivity that has a value along z direction different respect to x and y directions; the permeability K is considered only along the z direction, while along the x and y directions the permeability is null. The PCM inside the porous model is part of the solid phase, because the liquid PCM after the melting does not have any movements because it is blocked inside the cavity closed. Moreover, Pal and Joshi [13] established that in a honeycomb structure the natural convection of the PCM is neglected. The permeability K does not be affected by the PCM, because it depends only by the dynamic effects of the porous media. The permeability K is calculated by the analysis of Bejan [14] where the average velocity u_{avg} in a single parallel plate channel with is:

$$u_{\text{avg}} = \frac{H^2}{12\mu_f} \left(\frac{\Delta P}{L} \right) \quad (7)$$

Δp is the pressure drop along the channel, L is the channel length and μ_f is the dynamic viscosity of the air. Given that a porous medium obey to the Darcy law for low values of Reynolds:

$$\frac{\Delta p}{L} = \frac{\mu_f}{K} u_D \quad (8)$$

where u_D is Darcy velocity, the relation between Darcy velocity and average velocity is $u_D = \varepsilon u_{\text{avg}}$. Then the permeability K for a single channel is [14]:

$$K = \frac{\varepsilon H^2}{12} \quad (9)$$

The permeability for different CPL is:

$$K = \frac{\varepsilon H_n^2}{12} \quad (10)$$

The direct model is a conjugate heat transfer problem and it is compared with a porous model created with the following hypothesis:

- Natural convection is neglected in the whole system. This is consistent for the PCM because the enclosure cell is very small and the phenomenon does not appear. [13]
- The porous model is anisotropic. Along the x and y direction the thermal properties have different values.
- The Darcy law (eq. 8) is employed to describe the dynamic behavior of the porous media along the x

direction. In the other direction the permeability is null.

- The Local Thermal Equilibrium (LTE) model is adopted to simulate the heat exchange between the fluid zone and the solid zone in the porous model. The fluid zone is the air and the PCM is treated as a solid zone in the porous model.
- The solid phase in the porous model takes into account the behavior of the PCM and an unique solid temperature function T_s is defined.

The governing equations for 2D porous model are presented in Cartesian coordinates:

Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (11)$$

X-momentum equation

$$\begin{aligned} \rho_f \left(\frac{1}{\varepsilon} \frac{\partial u}{\partial t} + \frac{u}{\varepsilon^2} \frac{\partial u}{\partial x} + \frac{v}{\varepsilon^2} \frac{\partial u}{\partial y} \right) = \\ = -\frac{\partial p}{\partial x} + \frac{1}{\varepsilon} \left[\frac{\partial}{\partial x} \left(\mu_f \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu_f \frac{\partial u}{\partial y} \right) \right] - \frac{\mu_f}{K_x} u \end{aligned} \quad (12)$$

Y-momentum equation

$$\begin{aligned} \rho_f \left(\frac{1}{\varepsilon} \frac{\partial v}{\partial t} + \frac{u}{\varepsilon^2} \frac{\partial v}{\partial x} + \frac{v}{\varepsilon^2} \frac{\partial v}{\partial y} \right) = \\ = -\frac{\partial p}{\partial y} + \frac{1}{\varepsilon} \left[\frac{\partial}{\partial x} \left(\mu_f \frac{\partial v}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu_f \frac{\partial v}{\partial y} \right) \right] - \frac{\mu_f}{K_y} v \end{aligned} \quad (13)$$

Energy equation for the porous media in the case local thermal equilibrium (LTE), $T_s = T_f = T_{PCM} = T_{cord} = T$ is [15]:

$$\begin{aligned} (\rho c)_{eff} \frac{\partial T}{\partial t} + \rho_f c_{p,f} \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = \\ = \frac{\partial}{\partial x} \left(k_x \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k_y \frac{\partial T}{\partial y} \right) - \varepsilon \rho_{PCM} H_L \frac{\partial \beta}{\partial t} \end{aligned} \quad (14)$$

The component u, v are the air velocity inside the porous model and x, y are the cartesian coordinates, ρ_f is the air density, ε is the porosity of the air (fluid phase), p is the relative pressure, μ_f is the air dynamic viscosity, K_x and K_y are the permeability components of the porous zone. $C_{p,f}$ and k_f are respectively the air specific heat and air thermal conductivity, H_L is the latent heat of fusion. In this work the permeability component K_y is equal zero and the permeability component K_x is evaluated with equation (10).

The effective heat transfer capacity is:

$$(\rho c)_{eff} = \varepsilon (\rho c_p)_f + \varepsilon (\rho c_p)_{PCM} + (1 - 2\varepsilon) (\rho c_p)_{cord} \quad (15)$$

For the given geometric configuration, the fluid and the solid phases are in parallel for heat conduction in the longitudinal direction and in series for heat conduction in the transverse direction. Hence, the component of the equivalent thermal conductivity of porous media are [15, 16]:

- Parallel for heat conduction in the longitudinal direction (x-direction):

$$k_x = \varepsilon k_f + \varepsilon k_{pcm} + (1 - 2\varepsilon) k_{cord} + k_d \quad (16)$$

- Serial for heat conduction in the transversal direction (y-direction):

$$k_y = \frac{k_{cord} k_{PCM} k_f}{\varepsilon k_{PCM} k_{cord} + \varepsilon k_f k_{cord} + (1 - 2\varepsilon) k_{PCM} k_f} \quad (17)$$

where k_d is thermal dispersion. The thermal dispersion for parallel plates configuration result [17]:

$$\frac{k_d}{k_f} = \frac{3\varepsilon}{140} Pe^2 \quad (18)$$

with $Pe = u_{avg} H / \alpha_f$ is Peclet number of the channel.

In the equation (14), the liquid fraction β depends by the temperature T and the last term describes the melting of the PCM. Two different model will be studied and then compared. The first model is a direct simulation of the parallel plates system for different Channel per Unit Length (CPL = 2, 4, 8, 16 and 32) at the same porosity. The main problem of the direct model is related to the increment of the nodes number when the CPL is higher. The second model is the parallel plates system as a porous media for 2,4,8,16 and 32 CPL at the same porosity.

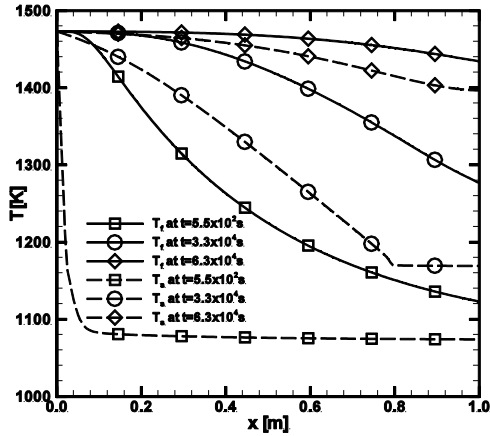
NUMERICAL PROCEDURE

The commercial CFD code Ansys-Fluent 17.2 is employed to solve the governing equations. A grid dependence test is accomplished to choose the best grid that represents a good compromise between the computational time with the solution accuracy. Three different grids are tested in the investigated geometry, $n_x \times n_y$ equal to 150 x 15, 300 x 30 and 600 x 60. The volumetric average temperature is monitored in the charging phase in the first cycle with a mass flow rate equal to 0.05 kg s⁻¹ m⁻². The maximum variation of the average temperature value when the number of the nodes are 300 x 30 with respect to the value obtained with the finest grid, 600 x 60, was 0.1%. The mesh 300 x 30 is employed in this investigation because it ensures a good compromise between computational time and accuracy requirements.

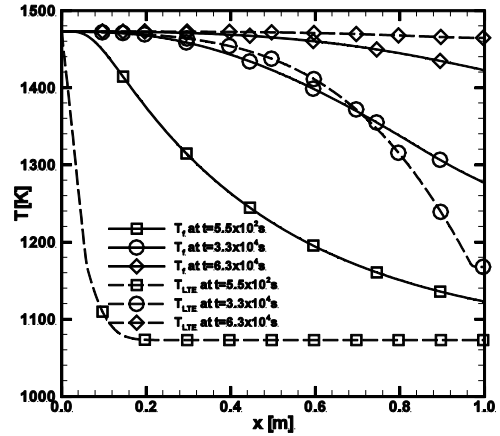
A transient analysis is made with a time step size of 1 s. For the pressure-velocity coupling the SIMPLE algorithm is employed, PRESTO algorithm is used for the pressure calculation, for energy and momentum equation the second order upwind scheme is used. To solve equations (14) the apparent heat capacity model [18] is implemented, in order to simulate the melting of the PCM. The inlet air temperature is 1473.15 K and the initial temperature of the system is 1073.15 K.

RESULTS AND DISCUSSION

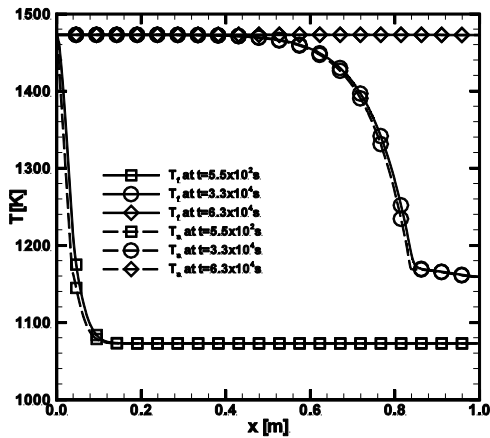
The results are presented in term of melting time, average temperature as function of time, temperature profiles for different CPL. In all case considered the a mass flow rate equal to 0.05 kg s⁻¹ m⁻² and the porosity value is 0.4. Figure 3 are showed the variation of temperature along the x direction in the centerline for parallel plates system in direct model for 4 and 32 CPL at different instants of time.



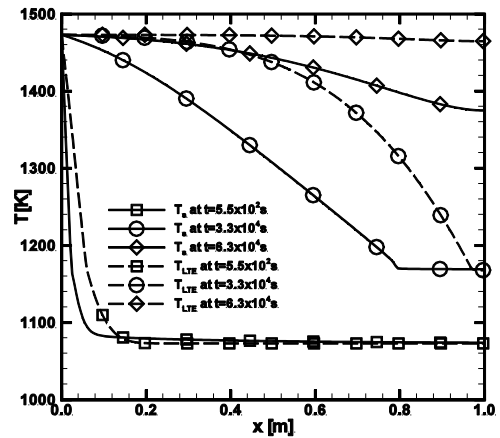
(a)



(a)



(b)



(b)

Figure 3. Air and PCM temperature along x direction in the parallel plates at different instants of time in direct model: (a) CPL=2 and (b) CPL=32.

Figure 4. Comparison between direct model and porous media model with CPL=4: (a) Air temperature in direct model and temperature in the LTE model, (b) PCM temperature in direct model and temperature in the LTE model.

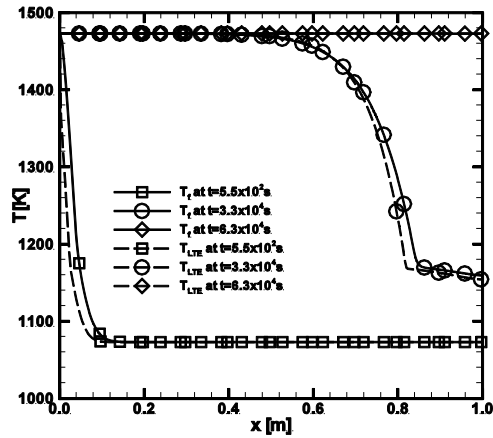
The temperature profiles in the direct model are evaluated along the centerline of the channel crossed by the air and the cavity closed filled with PCM.

In the figure 3a are reported the air and PCM temperature profiles along the x axis for different instants of time at 4 CPL. It is possible to see that the profile between the air and PCM present much differences at beginning of time but there are similar for larger instants of time. In the figure 3b the same profile are reported at 32 CPL. In this case the profiles for air and PCM are overlapped for each instant of time. Therefore, the local thermal equilibrium is valid at higher CPL.

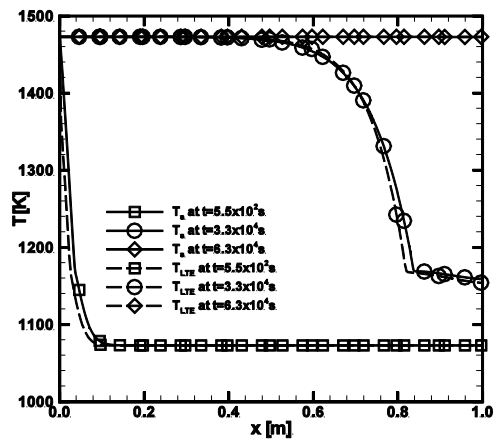
In figure 4 and 5 are depicted a comparison between the LTE (Local Thermal Equilibrium) porous model and the direct model for air and PCM temperature profiles. At CPL equals 2 the models presents some differences (figure 4a and 4c), but there is no difference for CPL equals 4.

CONCLUSIONS

A parallel plates channel system with PCM as thermal energy storage was numerically studied. A comparison between a direct parallel plates system and an analogous porous medium system was accomplished for different channels per unit of length (CPL). The local thermal equilibrium assumption for energy equation of porous medium model was adopted. Results in terms of temperature profiles of centerline as a function of time allowed the comparison between the direct and porous medium models. The results showed that for low CPL values (<4) the two models present some significant differences whereas these differences between direct and porous medium models become very small and they are smaller as greater CPL is. A more appropriate model, for small CPL values, in terms of porous medium could be under the local thermal non-equilibrium assumption. Some interesting developments are



(a)



(b)

Figure 5. Comparison between direct model and porous media model with $CPL=32$: (a) Air temperature in direct model and temperature in the LTE model, (b) PCM temperature in direct model and temperature in the LTE model.

related to simulate a thermal energy storage for a finite number of charging and discharging cycles and removing the adiabatic constraint allowing the heat transfer toward the external ambient.

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