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Numerical investigation of PCM solidification in a finned rectangular heat exchanger including natural convection

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ABSTRACT

The thermal energy storage got a significant role in the solar energy conservation to expand its use over time. To exploit solar energy continuously, we require a storage energy system. Phase Change Material (PCM) used in this kind of systems in order to store a great amount of thermal energy. This work concerns the solidification in the presence of the natural convection of a rectangular phase change material exposed to a cold airflow along the conducting sidewalls. With this intention, the first stage includes the presentation of a numerical model based on the conservation equations, treated by finite volume method. This numerical approach aims to follow the evolution of the various parameters characterizing the phenomenon of phase change (liquid-solid interface and solid fraction) during all the processes of solidification as well as the temperature and the velocity distribution in the PCM storage systems. Moreover, this study presents the transient evolution of the longitudinal air temperature profiles. It also presents the effect of fins number on heat transfer enhancement and the temporal evolution of the solidification front. The advantage of this study is to find a solution to extract the maximum of heat during the solidification of the PCM. This solution is to take into account the natural convection and also the addition of the fins.

1. Introduction

The development of 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. There are three modes of TES: Sensible Heat Storage (SHS), Latent Heat Storage (LHS), and Bond Energy Storage (BES) [1]. Latent Heat Thermal Energy Storage (LHTES) by PCMs is the most preferable storage techniques thanks to its important energy storage density and high isothermal storage process [2,3].

The use of PCMs to increase the thermal storage capacity in a heat exchanger presumes benefitting from the economic phenomena that occur at liquid-solid or solid-liquid phase changes. These phenomena involve the release of a large amount of heat during the solidification of a material and, reversely, the absorption of the same heat amount during the melting of the material.

Over the last twenty years, various authors [4–29] have been working to design the best active systems of the Heat Transfer Fluid (HTF)-PCM heat exchanger type. Active systems are systems in which the circulation of the HTF fluid in the exchanger is triggered by a mechanical element (ventilator, pump, etc.). For this purpose, Rostamizadeh et al. [10] have developed a one-dimensional mathematical model based on an enthalpy formulation for the study of the effect of the thickness of a PCM single-layer plate on the temperature distribution during the PCM melting, and on the temporal evolution of the liquefaction front. They have shown that the optimal thickness of the PCM is 5 mm, and the PCM mass and the melting time are in a proportionality relationship.

In the same context, Zivkovic et al. [11] have studied the accumulation of thermal energy in a chamber filled with PCMs, of rectangular shape and heated by the circulation of a current of air parallel to their large faces. The chamber is supposed to be isolated by its four lateral facets. The numerical results found by these authors have shown that the conduction in the PCM in the direction of the axis (oz) can be neglected when dealing with the storage modules of quite short length (less than 1 m). This is accurate since the convective transfer coefficient h_{conv} is low. Moreover, these authors have shown that for flat and thin containers, the effects of natural convection in liquid PCM can be ignored.

El Qarnia [12] and Brousseau et al. [13] have examined the thermal performance of a multilayer storage system. The storage

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Nomenclature

C_{air}	heat capacity of air ($J \cdot kg^{-1} \cdot °C^{-1}$)
C(T)	heat capacity per unit volume including the phase change ($J \cdot m^{-3} \cdot °C^{-1}$)
C_l	liquid phase heat capacity per unit volume ($J \cdot m^{-3} \cdot °C^{-1}$)
C_s	solid phase heat capacity per unit volume ($J \cdot m^{-3} \cdot °C^{-1}$)
dA	the elementary heat transfer exchange area (m^2)
dV	the representative elementary volume of the air duct (m^3)
f	liquid fraction
g	gravity acceleration ($m \cdot s^{-2}$)
h	convective heat transfer ($W \cdot m^{-2} \cdot °C^{-1}$)
L	latent heat ($J \cdot kg^{-1}$)
Re	Reynolds number
x, z	coordinates system
S_U, S_W	momentum source terms ($m \cdot s^{-2}$)
S_T	thermal source term in air equation ($W \cdot m^{-3}$)
T	temperature of PCM (°C)
T_0	inlet air temperature (°C)
T_{air}	temperature of air (°C)
T_{liq}	temperature of liquid interface (°C°C))
T_{sd}	temperature of solid interface (
T_{ref}	reference temperature of PCM (°C)
T_{PCM}	wall temperature in z section (°C)
U, W	velocity components of PCM ($m \cdot s^{-1}$)
W_{air}	air velocity ($m \cdot s^{-1}$)

Greek symbols

$\rho(T)$	density including the phase change ($kg \cdot m^{-3}$)
ρ_l	liquid phase density ($kg \cdot m^{-3}$)
ρ_s	solid phase density ($kg \cdot m^{-3}$)
ρ_{air}	air density ($kg \cdot m^{-3}$)
λ_l	liquid phase heat conductivity of PCM ($W \cdot m^{-1} \cdot °C^{-1}$)
λ_s	solid phase heat conductivity of PCM ($W \cdot m^{-1} \cdot °C^{-1}$)
λ_{air}	heat conductivity of air ($W \cdot m^{-1} \cdot °C^{-1}$)
Δt	time step (s)
Δz	longitudinal step (m)

device is formed by the juxtaposition of several parallel flat PCM plates, constituting free spaces between them, in which the coolant (air for example) circulates. In their study, the effect of plate thickness and plate spacing was of particular interest.

Furthermore, Zalba et al. [14] have studied an air conditioning system of a room, consisting of a bulk of plates of δ thickness of PCM arranged to form canals for the passage of air. The analysis showed a significant effect of plate thickness during the solidification and liquefaction process.

Stritih [15] experimentally studied a PCM heat storage unit made up of a rectangular tank. The used PCM is paraffin with a melting point of 30 °C, which is suitable for thermal storage applications in buildings. These authors compared the thermo-convective behavior in the PCM during the melting and solidification phases. They demonstrated that during solidification, conduction is the dominant form of heat transfer. On the other hand, during the fusion, the natural convection has become substantial. They recommended the addition of PCM-side fins to improve conduction transfer, noting a 40% reduction in solidification time.

Seck et al. [18] have introduced a large study based on analytical, experimental and numerical approaches to control the evolution of the melting front of a paraffin plate exposed to sunshine. This study was conducted on a parallelepiped storage sensor. The problem is supposed to be purely conductive and one-dimensional. The comparison of the numerical and experimental results has shown that the more important the liquid phase becomes, the more marked discordance becomes. Hence, it seems that the thermo-convective phenomena play an essential role in the propagation kinetics of the fusion front.

Khalifa et al. [26] have presented numerical and experimental investigations on the thermal performance of LHTES systems, which use heat pipes for the generation of solar thermal power. These researchers have shown the benefits of added fins.

Recently, a numerical approach for modeling the PCM melting in a rectangular cavity is proposed by El Qarnia el al. [30]. The influence of different key parameters on the PCM-based heat sink's thermal performance is investigated through numerical simulations. They have developed two correlations, one for the appropriate melt fraction and the other for the maximum working time. The proposed approach can be useful for the design of PCM-based cooling systems.

The present work presents a numerical investigation of PCM solidification (paraffins C_{18} and RT27) in a finned rectangular PCM-air heat exchanger. The originality of this work lies in the fact of taking into account natural convection during the heat transfer phenomenon at the solidification of the PCM. The first stage includes the presentation of a numerical model based on the continuity, momentum, and thermal energy equations, treated by a finite volume method. The main objective of this numerical approach is the study of the effect of natural convection on the PCM solidification time and the impact of fins number on heat transfer enhancement. It also aims at investigating the temporal evolution of PCM solidification, and solidification front, as well as the longitudinal profiles of the HTF in duct.

2. Computational domain and mathematical formulation

In the present work, the studied system is a mass of several metal flat modules filled with PCM and arranged horizontally (Fig. 1). Each module is of a length L = 1.2 m and a height H = 4 cm. The quincunx montage of vertical fins of rectangular profile, on the upper and lower plates constituting the envelope of each PCM module, is dictated by the desire to improve the PCM-air heat exchange performance.

With respect to the HTF fluid, it is conveyed in the rectangular conduits composed of the free spaces between two successive PCM modules. Each conduit is divided into grooves by rectangular fins increasing the exchange surface between the HTF (air) and the upper and lower walls of the PCM tank, and thus contributing to a more intense exchange on the airside. These fins are arranged perpendicular to the plans of those immersing in the PCM. The grooves have straight sections of dimensions a = b = 4 cm.

The problem is two-dimensional (x-z). The computational domain corresponds to the surface framed in green[1] in Fig. 2. The mass flow rate of air is 0.08 $kg \cdot s^{-1}$. The fins are made up of aluminum with a constant thickness of 1 mm. The thermo-physical properties of the solid and liquid PCM and aluminum are listed in Table 1.

Four different geometry configurations have been studied. The first configuration treats a PCM-air heat exchanger of module without fins, and the other three configurations concern the same heat exchanger design with the addition of 3, 5 and 9 fins on the upper and lower plates constituting the envelope of each PCM module

[1] For interpretation of color in Fig. 2, the reader is referred to the web version of this article.